CRANFIELD UNIVERSITY

ANTONY ALDRIDGE

DOWNHOLE MANIPULATION SYSTEMS FOR ABRASIVE JET CUTTING IN OIL WELLS

A DESIGN STUDY

SCHOOL OF INDUSTRIAL AND MANUFACTURING SCIENCE

PhD. THESIS

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ACADEMIC SUPERVISOR : Dr R. ALLWOOD

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ABSTRACT

This project investigates the possibility of using abrasive water jets for cutting operations inside oil wells and presents the associated procedures and tools required. Through-tubing window cutting was identified as an application of particular interest to the oil industry. This involves cutting a hole in the well casing, below the production tubing, so that a new well can be directionally drilled through it and into the producing formation.

The study showed that through-tubing window cutting is a feasible application for abrasive water jets. The main findings are:

- a 2.5 m by 0.1 m window can be cut in 10 mm thick casing in approximately 12 hours. This assumes that the window is 3000 m from the surface and that a surface pressure of 595 bar is used.

- a manipulator has been designed in detail, based on a pressure balanced cylinder, which can move the nozzle smoothly and accurately over the complete surface area of the window. The maximum outer diameter is 53 mm and it would be deployed on 1½" coiled tubing. Test rigs have confirmed the operation of critical components of the design.

- several aspects of the design still need to be tested. A prototype manipulator should be built to demonstrate that it can repeatedly cut the same size of window and operate continuously for over 20 hours.

- the pressure balanced design principle is applicable to any abrasive water jet cutting manipulator in which the nozzle is perpendicular to the direction of the well. Two examples are perforation of short sections of casing and cutting around the circumference of the production tubing for removal.

- a specially designed cutting fluid with appropriate polymer additives must be used to give excellent solids carrying properties and low pressure loss. Their effect on cutting performance and hole cleaning efficiency needs to be tested.

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NOMENCLATURE

Unavoidably, some symbols are used for several expressions. Therefore the symbols are always defined with each set of equations.

The most common symbols used are:

Symbol	**Meaning**
A	cross sectional area
C_d	coefficient of discharge for nozzle
d	diameter
e	pipe wall roughness
E	Young's Modulus
f	friction factor or design stress
g	acceleration due to gravity
h	depth of operation
I	moment of inertia
ID	inner diameter
k_t	stress concentration factor
ℓ	actual length of column under load
ℓ_k	length of column under load, factored to account for the type of support
L	length
N	number of nozzles
OD	outer diameter
p	thread pitch
P	pressure or buckling load
ΔP	pressure loss or pressure drop
PS	parting speed
Q	volume flow rate of cutting fluid per nozzle
Re	Reynolds number
t	time or minimum length of thread
T	thickness
TS	traverse speed
u	fluid velocity
V	total volume flow rate of cutting fluid
W	width

Greek Symbols

σ	stress
ρ	fluid density
μ	fluid dynamic viscosity

Subscripts

N	nozzle
eff	effective
CT	coiled tubing
Ann	coiled tubing - casing annulus
Lim	limit
W	window
f	female
m	male
min	minimum
$crit$	critical

ABBREVIATIONS

API	American Petroleum Institute
AWJ	Abrasive Water Jet
AWJC	Abrasive Water Jet Cutting
BHA	Bottom Hole Assembly
BOP	Blow Out Preventer
CT	Coiled Tubing
DIAJET	Direct Injection Abrasive JETting*
ED	Emergency Disconnect
FPG	Formation Pressure Gradient
GRI	Gas Research Institute
HP	High Pressure
JPT	Journal of Petroleum Technology
SG	Specific Gravity
SPE	Society of Petroleum Engineers
UHP	Ultra High Pressure

* Abrasive water jet cutting equipment, patented and manufactured by DIAJET Limited.

GLOSSARY OF OIL FIELD TERMS USED

Back pressure	The pressure due to the static head of fluid in the well.
Build Rate	A term used in directional drilling to describe the rate of change of direction of the drill bit.
Coiled Tubing	Continuous steel tube available in lengths up to approximately 15,000 ft and outer diameters from 1" to 3.5". The tube is wound onto a reel.
No-go nipple	A restriction in the end of the production tubing to prevent tools falling into the well below. Certain types of measuring tools are designed to latch into the nipple.
Packer	An inflatable element which seals in the well bore.
Perforations	Holes made through the casing and into the formation to enable the reservoir fluids to enter the well.
Production Tubing	Jointed tubing inside the casing, up which the production fluids (oil, gas and water) flow to the surface. A packer is used at the bottom of the tubing to seal the production tubing-casing annulus.
Wireline	Electrical cable(s) installed inside the drill pipe or coiled tubing.

1.0 INTRODUCTION

1.1 Preamble

DIAJET Limited, a subsidiary of BHR Group, is continually looking for new applications for their abrasive water jet cutting system, DIAJET. An area which has shown considerable potential is in the petroleum industry, where it could possibly be used for a wide range of cutting operations inside oil wells. Of particular interest was using an abrasive water jet to cut a window in the well casing below the production tubing, at distances of over 3000 m from the surface.

Window cutting involves cutting a hole (window) in the side of the well casing so that a new well can be drilled through it and into the producing formation so that production can be increased. This is much cheaper than drilling a completely new well from the surface.

This PhD thesis presents a feasibility study for a Through-tubing Abrasive Water Jet Window Cutting System and proposes a detailed manipulator design to move the nozzles in order to cut the window.

The oil industry has considerable experience of using abrasive and plain water jets. However, successful operation within oil wells has been limited due to a number of constraints. These include:

a) the maximum operating pressure and flow rate which can be achieved at the nozzle. These are restricted by the dimensions and pressure ratings of the connecting pipework.

b) providing a suitable carrier fluid to transport the cuttings back to the surface.

c) providing a water jet system which is proven, as far as practicable, to be reliable and economical. Oil companies are reluctant to try new ideas because the expense involved with failures can be astronomical. They will usually prefer to 'make do' with existing technology.

d) the extreme environment (pressure, temperature and corrosiveness) inside an oil well and the difficulties of conducting operations thousands of metres from the surface.

1.2 Aims of the Project

Based on these problems and many more which will be identified during the first four chapters, the following areas of work were identified:

i) To identify the main advantages and disadvantages of abrasive water jet cutting inside oil wells by reviewing their use in the oil industry.

ii) To identify the critical operating parameters for cutting a window, such as the minimum allowable time to cut the window. The minimum abrasive water jet requirements can then be determined (pressure, volume, traverse speed).

iii) To determine methods for conveying the cutting fluid, at the required pressures and volumes, to the nozzle(s).

iv) To demonstrate that, in theory, window cutting is a feasible downhole application for an abrasive jet cutting system.

v) To investigate the design of a manipulator which can position and move the nozzle(s) as required to conduct the selected cutting operation.

vi) To investigate and incorporate sensors into the system as required. For example, sensors to detect that acceptable cuts have been achieved.

vii) From the results and conclusions of this specific investigation, consider other cutting applications which may be feasible.

1.3 Thesis Outline

Chapters 2, 3 and 4 provide the necessary background for this thesis and discuss relevant previous work in the area of downhole abrasive water jet cutting, thus addressing aims (i) and (ii) from the above list. The remainder of the thesis details my own study on an abrasive water jet window cutting system. The thesis comprises of the following chapters:

Chapter 2	An overview of abrasive water jet cutting and its advantages and limitations for cutting operations inside oil wells.
Chapter 3	A literature review of downhole applications of abrasive water jets and the advantages and disadvantages of using them.
Chapter 4	A review of window cutting techniques and the potential for an abrasive water jet window cutting tool.
Chapter 5	A feasibility study for an abrasive water jet window cutting tool. This includes cutting results and a mathematical model to establish the theoretical time to cut a window.
Chapter 6	A Specification for an abrasive water jet window cutting tool, summarising all of the main points identified in Chapters 3, 4 and 5.
Chapter 7	This presents the development of an abrasive water jet window cutting manipulator and the results of test rigs designed to demonstrate the feasibility of the critical features of the design.
Chapter 8	In conjunction with Appendix G, this presents the final manipulator design.
Chapter 9	Conclusions and future work.

2.0 ABRASIVE WATER JET CUTTING

2.1 Introduction

DIAJET Limited is an internationally renowned centre for the research and development of abrasive water jet cutting (AWJC) tools.

Legislation is leading to the restriction of some cutting techniques, and the introduction of new, stronger and tougher materials require new tools to cut them. AWJC is a new technology which can satisfy a number of these cutting requirements (Ref 1 and 2). The main advantages of AWJC include:

1) the method will cut any material including metals, ceramics, glass and polymers.

2) the cutting process is cold and consequently there are no heat affected zones which change material properties.

3) this cold nature makes it ideal for use in explosive or flammable environments.

4) jet reaction forces are low, enabling accurate manipulation by robots and multidirectional cutting.

5) minimal or no dust is produced.

6) when using small diameter nozzles, for example 0.2 mm, kerf widths (width of the cut) comparable to the diameter of the nozzle can be achieved and waste material is therefore minimised. Also, by using small nozzles, noise is reduced.

A detailed description of applications for plain water jets and abrasive water jets is given in Reference 3.

For any type of jet cutting system, although the performance of the system is extremely important, the ability to accurately control the position of the nozzle or the work piece is equally important. Consequently the development of advanced robotics and numerically controlled cutting tables have started to enable the full potential of these systems to be realised.

2.2 Entrainment and Direct Injection Type Abrasive Jet Cutting Systems

Two types of abrasive cutting systems are available, the more common entrainment systems and direct injection systems, such as 'DIAJET'.

2.2.1 The Entrainment System

The entrainment system is based on high pressure (HP) water being accelerated through a nozzle into a large volume mixing chamber. The high velocity water generates a suction pressure in the mixing chamber which causes abrasive to be sucked into the chamber and entrained into the flow. See Figure 2.2.1. The subsequent mixture is then focused through a larger nozzle and directed at the target.

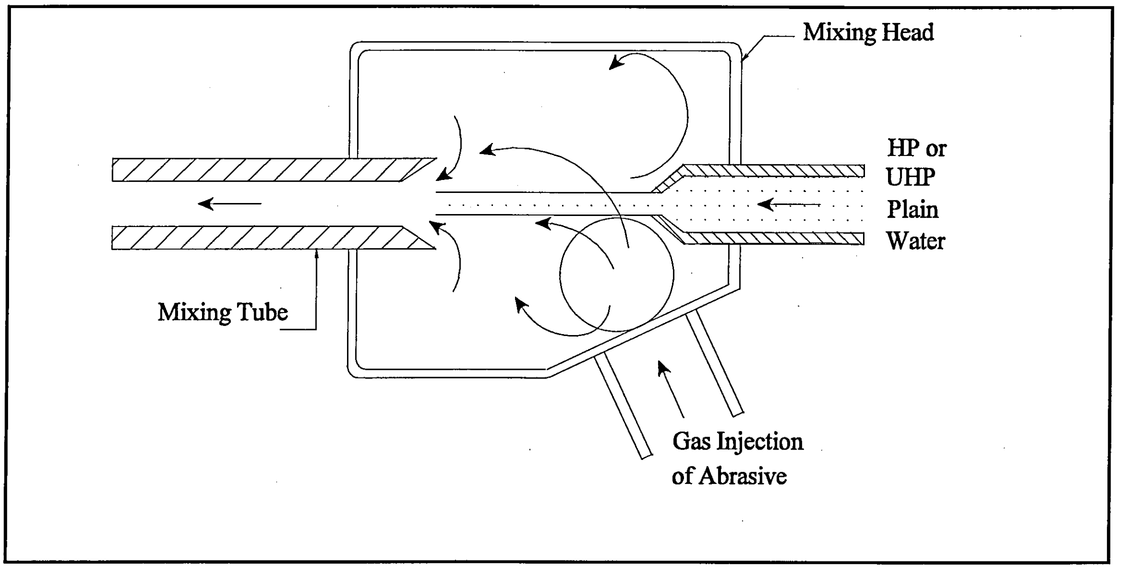


FIGURE 2.2.1 AN ENTRAINMENT HEAD

Although this system is very simple, the action of accelerating the abrasive from a low velocity in the suction line to the high velocities in the water jet is inefficient and deteriorates as the abrasive mass flow increases. The final jet is not very homogenous, which results in uneven wear of the mixing tube and focusing nozzle and necessitates the early replacement of the nozzle.

2.2.2 Direct Injection

The direct injection DIAJET system is based upon having the abrasive in a slurry form, already at system pressure. The pressurised water-abrasive mixture is then accelerated through a specially designed nozzle as a high velocity, homogenous jet. By varying the quantity of high pressure water within the slurry, the abrasive concentration can be controlled (see Figure 2.2.2).

For the same input power, the DIAJET system has been shown to achieve cutting rates four times better than comparable entrainment systems. However, because of its complexity (requirement for pressure vessels etc), commercial units have been limited to a maximum nozzle pressure of 690 bar (although experimental systems are available up to 4000 bar) compared to 3000-4000 bar for an entrainment system. Consequently, to achieve the same cutting performance as a high pressure entrainment system, the DIAJET system must use a larger nozzle. More abrasive and water is therefore used, which results in higher operating costs.

As an example of the DIAJET cutting performance, a 2.0 mm diameter nozzle will cut 16 mm of 304L stainless steel at a pressure of 345 bar, traverse speed of 125 mm/min, abrasive concentration of 9% by weight and a stand-off of 10 mm.

Note that in abrasive water jetting the usual units of measure are bar for pressure and litres/min for volume flow rate.

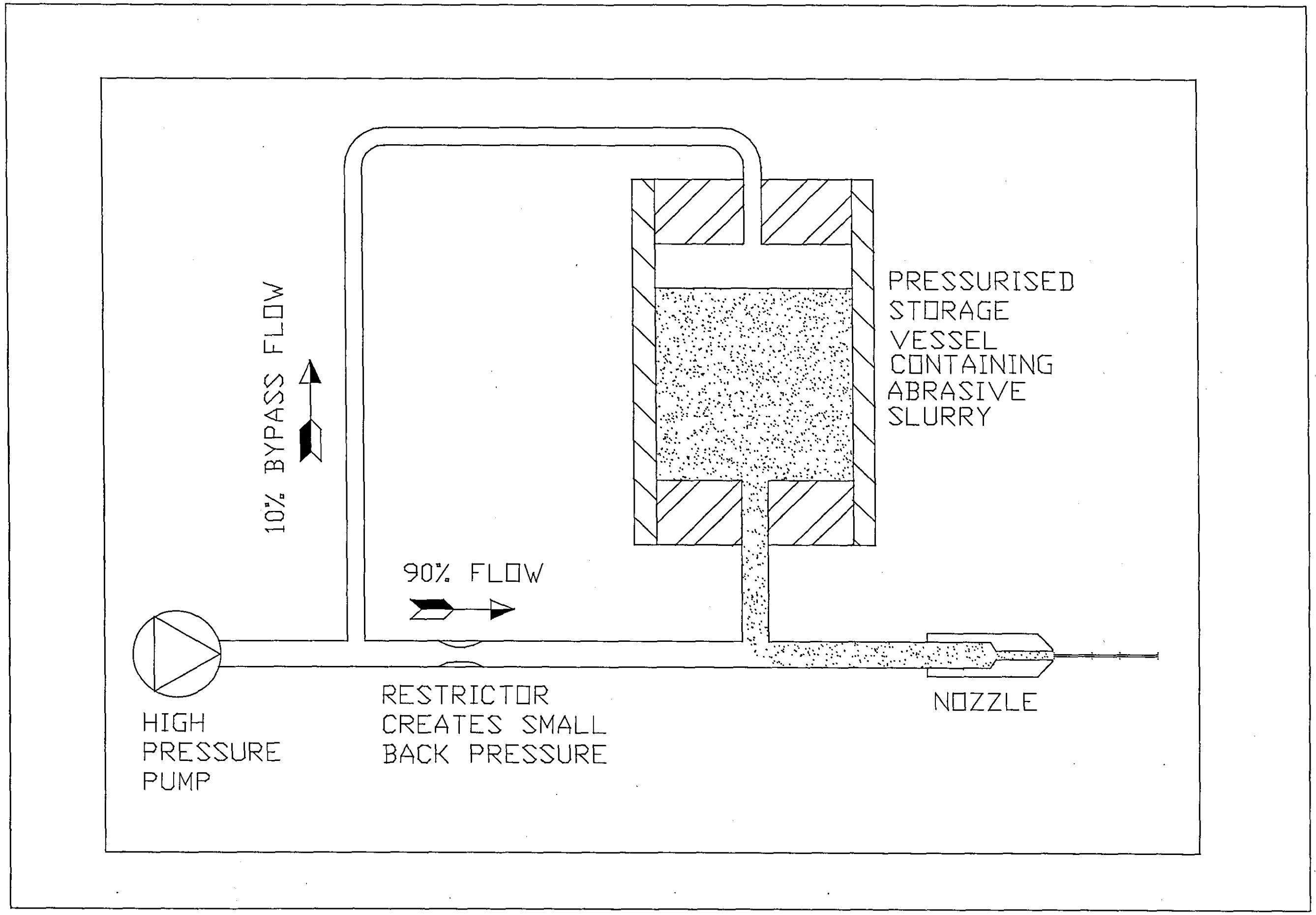


FIGURE 2.2.2 SIMPLIFIED DIAJET CIRCUIT

2.3 The Parameters That Affect the Performance of an Abrasive Water Jet Cutting System

The basic parameters which influence the performance of an abrasive water jet cutting system are:

Jet pressure	Nozzle diameter
Jet traverse speed	Number of passes
Stand-off distance	Abrasive feed rate
Abrasive size	Abrasive properties
Workpiece properties	

These parameters are considered in more detail below. However, it is important to note that it is the combination of each of these factors which determines the final cutting performance.

1) Jet Pressure

All materials demonstrate a threshold critical pressure below which no cutting can occur (Ref 2). This threshold reduces with increasing abrasive flow rate and increases with increasing traverse speed.

2) Nozzle Diameter

Small nozzles use lower volumes of water and abrasive, cut narrower slots, concentrating the available jet power onto less target material, and consequently are more efficient at utilizing the hydraulic power available at the nozzle (Ref 2 and 4). However, the abrasive size must be reduced which may result in a reduction in the absolute cutting performance, although the cut quality and surface finish may be improved.

The governing equations for fluid flow through a nozzle are given in Appendix A.

3) Traverse Speed

If the nozzle is held stationary, cleaning of the erosion zone becomes difficult as the exhaust stream has to overcome the coaxial incident jet. This results in the hindrance of the cutting process as the velocity of the incident particles is slowed (Figure 2.3.1.a)(Ref 5).

By moving the jet at a sufficient speed, there will be a smooth transition from incident jet to exhaust stream and the cutting performance will be improved dramatically (Figure 2.3.1.b). If the traverse speed is then further increased the cutting performance will start to deteriorate, the amount of material required to be removed becoming too great.

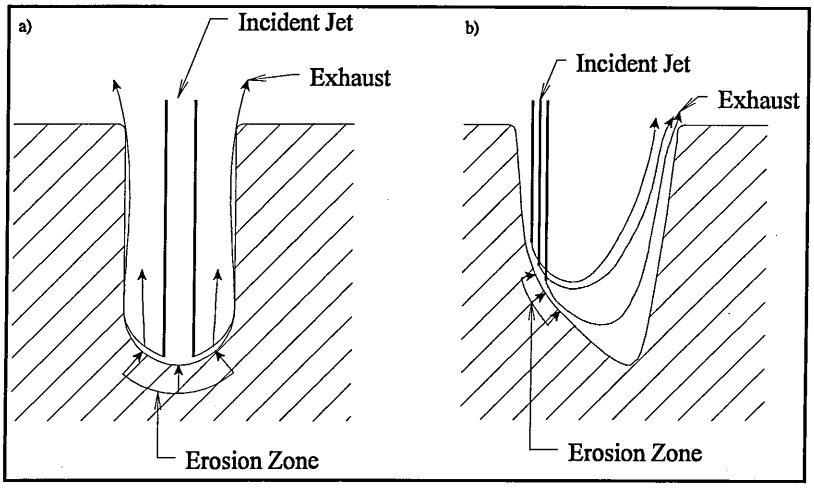


FIGURE 2.3.1 COMPARISON OF THE MATERIAL MOVING MECHANISMS: a) for a stationary jet and b) for a moving jet.

4) Stand-Off Distance

Increasing the distance from the nozzle to the target results in an increased jet diameter at the target. Therefore, the depth of cut is generally observed to decrease but the width of cut increases. At the optimum stand-off distance (usually expressed as a function of the nozzle diameter) the maximum volume of material can be removed.

5) Number of Passes

When the jet is redirected over the target for a second cut, the depth of cut achieved will depend on the effectiveness of the jet at an effectively increased stand-off distance. Generally an optimum depth of cut can be achieved with the correct number of passes at the best traverse speed. Increasing the number of traverses beyond this optimum will only give minor increases in the depth of cut until the limit is achieved.

6) Abrasive Flow Rate

Increasing the abrasive flow rate will increase the number of particles impacting the target material, increasing the depth of cut or the cutting speed that can be achieved. However, above a critical concentration the particles start to interfere with each other, reducing their cutting effectiveness.

7) Abrasive Materials

Abrasive materials ranging from common silica sand to garnet and copper slag can be used. In general, particularly for harder target materials, the higher the abrasive hardness the better depth of cut that can be achieved. This must be balanced against the higher purchase cost and increased wear of the system components associated with the harder abrasives.

8) Abrasive Particle Size

In order to ensure nozzle blockage does not occur, the abrasive is sieved to between 30% and 50% of the nozzle diameter.

9) Target Material Properties

Generally the depth of cut decreases as the target material hardness increases. The cutting performance will also depend on whether the material is ductile or brittle. Ductile or brittle erosion is discussed in detail by Fairhurst (Ref 6) and is mainly based on the work of Finnie and Bitter.

The complex interaction of all of these parameters makes it very difficult to isolate the influence of one parameter on the cutting performance. To obtain the required cut it is often only feasible to alter one or two of the parameters, which restricts the final cutting performance that can be achieved. For example, for remote operations ensuring a minimum stand-off distance might be a problem.

2.4 Cutting Models

DIAJET use a cutting model specific to DIAJET cutting written by Claffey (Ref 7). This model applies to cutting ductile materials only. Although it models the trends very well (depth of cut for different nozzle diameters or pressures) the exact prediction can vary by as much as 50%. This is believed to be due to several factors, the most important being the difficulty of modelling the work hardening of the metal as it is cut. In most cases, an initial linear cut is adequate for calibrating the model.

For constant abrasive flow rates and a nozzle pressure drop of less than 500 bar, the following simple equation can be applied:

$$Depth\,of\,Cut \;\alpha\; \frac{(Nozzle\,Pressure)^{1.5}\,.\,(Nozzle\,Diameter)}{(Traverse\,Speed)}$$

Therefore, from one initial cut the cutting performance for the same material but different configuration can be predicted.

2.5 The Suitability of Entrainment Systems and DIAJET Systems For Downhole Operations

Oil wells are usually filled with fluid, either formation fluid such as water, brine or oil, or drilling fluids. The abrasive water jet will therefore be submerged when it is cutting. Alberts and Hashish (Ref 8) describe work that they conducted to investigate the suitability of entrainment-type systems and direct injection systems for use in subsea environments.

One major disadvantage of an entrainment system for this purpose, is the requirement of two supply lines to the cutting head, one to convey the high pressure water and one to convey the abrasive. This increases the cost and makes the manipulation of the nozzle more complicated.

The abrasive for an entrainment system can either be carried to the cutting head by air, in which case the abrasive is dry, or in a low pressure slurry (Ref 8 and 9). Haferkamp (Ref 10) describes tests conducted using these two types of entrainment systems. Cutting was conducted submerged, in a special pressure chamber to simulate the higher ambient pressures experienced subsea. The dry abrasive feed was found to be influenced by ambient pressure, the compression of the supply air resulting in surrounding water flowing into the focusing nozzle and hence reducing the effectiveness of the exiting jet. The slurry arrangement, although found not to be too sensitive to ambient pressure, was inefficient because both the abrasive and the water carrying it had to be accelerated by the high velocity water jet.

References 8, 9 and 10 all conclude that for subsea applications a direct injection system is a more effective cutting device, although the cutting performance was found to deteriorate with ambient pressure in a similar way to the entrainment system with the slurry feed. However, the main disadvantage of the direct injection systems is the larger quantities of abrasive and water that are required because of the lower operating pressures, as mentioned in Section 2.2.

All of these studies confirm that the DIAJET system should be suitable for providing the cutting fluid required for a downhole cutting tool.

The use of abrasive water jets in the oil industry and some of the problems associated with them becoming a widespread technology is discussed in Chapter 3.

2.6 Cutting Inside Oil Wells

For cutting inside oil wells, two further parameters need to be considered:

1) the effect of back pressure on the jet, due to the static head of fluid in the well.

2) the effect of the jet being submerged, usually in a viscous fluid.

2.6.1 Effect of Back Pressure

Grove (Ref 11) found that for a constant pressure difference across the nozzle, the depth of cut in rock reduces as the back pressure increases. The depth of cut decreases most markedly for back pressures up to 100 to 170 bar, which is believed to be associated with the suppression of jet cavitation (Ref 8) and the possible change in the mechanical properties of the rock. For back pressures of 300 bar, the depth of cut was found to reduce to 40% to 60% of the depth achieved at normal ambient pressure, although the general trend, shown in Figure 2.6.1, is a flattening of the curve as the back pressure increases.

Note that jet cavitation is usually considered to be a secondary erosion effect in abrasive water jet cutting, but it is encouraged in high pressure plane water jets to improve their cutting performance.

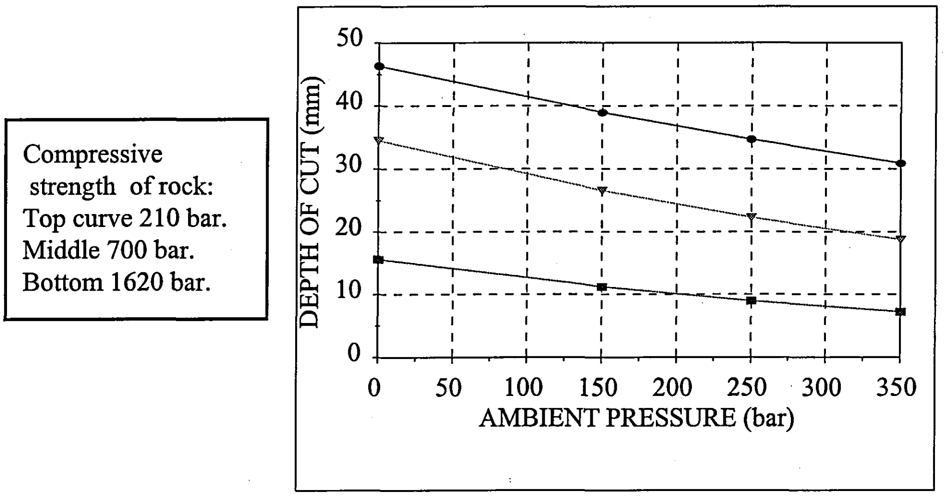


FIGURE 2.6.1 THE EFFECT OF BACK PRESSURE ON THE DEPTH OF CUT ON ROCKS

Little similar information is available on the cutting of steel. Domann (Ref 12) conducted some piercing trials on steel using a DIAJET (2.4 mm nozzle at 120 bar) with the steel sample held in a pressure chamber. He found that the erosion rate decreased significantly in water depths of between 50 m and 100 m (back pressure of 5 to 10 bar) and then stayed roughly constant down to 500 m depth (50 bar).

Wakefield (Ref 13) detailed some cutting results he obtained when using a DIAJET (2.8 mm nozzle at 133 and 292 bar) to cut steel in a hyperbaric vessel. He found that the depth of cut reduced by between 17% and 29% as the back pressure increased from 3 to 10 bar. As the back pressure was increased further, a slight increase in the depth of cut was observed, for which no explanation was given.

Kolle (Ref 14) gives a method to calculate the back pressure required to suppress cavitation (See Appendix B). Table 2.6.1 details the expected back pressure required to suppress cavitation for the examples given above.

TABLE 2.6.1 TO SHOW THE MINIMUM BACK PRESSURE REQUIRED TO SUPPRESS CAVITATION FOR THE EXAMPLES DISCUSSED.

Nozzle Diameter (mm)	Material Cut.	Nozzle Pressure (bar)	Measured Back Pressure at which Performance is Reduced. (bar)	Minimum Back Pressure to Suppress Cavitation. (bar)
1.5	Rock	350	100 - 170	70
1.5	Rock	500	100 - 170	100
1.5	Rock	600	100 - 170	120
1.5	Rock	700	100 - 170	140
2.4	Steel	120	5 - 10	24
2.8	Steel	133	10	27
2.8	Steel	292	10	58

The Stand-off distance is assumed to be small for all of the examples (less than 5 nozzle diameters).
The minimum back pressure to suppress cavitation = 0.2 * Nozzle Pressure.
(See Appendix B)

The measured results for steel cutting indicate that the cutting performance significantly reduces at back pressures lower than the calculated back pressure to suppress cavitation. For cutting rock, the pressure range at which the performance was observed to reduce is too large to compare with the back pressure required to suppress cavitation.

No relationship between the depth of cut and the back pressure has been found. Although Wakefield measured a 30% reduction in performance, more information is required.

To ensure complete cutting with the nozzle submerged and at back pressure, a reduction in performance of 50%, as measured on rock, will be assumed. Therefore, if 10 mm thick steel has to be cut, the nozzle size, pressure and traverse speed will be based on being able to cut a minimum of 20 mm.

2.6.2 Effect of Submerging the Jet

Investigations into the performance of jets submerged in water (Refs 8, 9 and 10) has shown that the cutting performance is more sensitive to the stand-off distance than when cutting in air. Blevin shows that for a submerged, plain water jet the exit velocity is constant up to 5 nozzle diameters away from the nozzle (Ref 15). The abrasive particle will either still be accelerating in this region or will be at a constant velocity also, so the nozzle should always be held less than 5 nozzle diameters away from the target.

More information is needed on the effect of submergence in viscous fluids. Polymer additives may be required in the cutting fluid to maintain jet coherence and cutting efficiency, although after several hours of cutting, when the surrounding environment will be filled with polymer, their effect will be limited.

3.0 DOWNHOLE CUTTING APPLICATIONS FOR ABRASIVE WATER JETS

The potential use of water jets and abrasive water jets in the oil industry has been recognized for many years. Ousterhout dates the first commercial use of a sand-laden fluid for perforating in 1939 (Ref 16). A review of literature on the use of abrasive water jets in oil wells uncovered papers which spanned over 30 years. Surprisingly, however, for each particular application, the reports contain very similar details and conclude that in general an abrasive jet cutting tool is too slow.

This highlights the common problems faced by abrasive water jet technology. In principle they offer many advantages and, although used for specific applications, the cost of developing them further, the operational difficulties associated with them and the speed at which alternative tools have become available seems to have limited their use.

This chapter discusses some of the past and current oil well applications for which abrasive water jet systems have been considered, some of the problems that have been encountered and the advantages and disadvantages of using them. This provides the necessary background information which must be taken into account when the task of cutting a window with an abrasive water jet is considered.

In order to understand these applications more fully, background information relating to specific oil well operations is given. Some comparative cutting trials that I conducted at DIAJET Limited are also described.

Downhole applications for abrasive water jet cutting systems include:

- Drilling
- Perforating and fracture initiation
- Descaling
- Cutting and decommissioning

The use of plain water jets downhole is an enormous subject alone. Only specifically relevant applications of plain water jetting have therefore been included.

Note: most of the units quoted in this section are Imperial because these are traditionally used in the oil industry.

3.1 Abrasive Water Jet Drilling

In 1993 the Gas Research Institute (Ref 17) quoted that on average, for wells drilled deeper than 15000 ft (5000 m), 49% of rig time was spent drilling. Consequently, considerable effort has been spent trying to improve drilling performance. Conventional rotary drilling has progressed rapidly, particularly with the design of the polycrystalline diamond bits (PDC bits), such that rates of penetration as high as 300 ft/hr (0.025 m/s) can now be reliably achieved, although rates of 20 to 100 ft/hr are more typical. In some wells, the rate of penetration has to be restricted because the volume of cuttings generated cannot be efficiently removed from the well.

Over the last 30 years other techniques have been considered, including the use of high pressure water jets and abrasive water jets. A detailed review of the research conducted on abrasive jet drilling is given by Maurer in his book Advanced Drilling Techniques (Ref 18). Only a few specific examples will therefore be discussed in detail.

Jet assisted mechanical drills are the most commonly described arrangement in which the jets, either abrasive or high pressure plain water, cut into the rock ahead of a drill bit. The bit can then easily remove the remaining narrow ridges of rock. These systems can significantly increase the rate of penetration and require lower weights to be applied to the bit, which reduces bit wear and consequently fewer trips are needed to change bits.

3.1.1 The Gulf Oil Project

Gulf Oil in the early 70's conducted the most extensive research program on the use of abrasive jet assisted mechanical drills. Initially sand was used for the abrasive. However, the particles broke down after only one pass, preventing their reuse. Steel shot was then used which could be recycled. The drilling mud, which was pumped through modified drill pipe, was specially designed to suspend the steel shot and to minimise friction loss. It is very important that the steel shot does not settle if circulation is stopped, such as when the nozzle blocks, otherwise removal of the drill string would be difficult. High pressure bits containing 20, $\frac{1}{8}$" (3.175 mm) jet nozzles were supplied with abrasive fluid at 10,000 psi.

Drilling results showed that a 4 to 20 fold increase in speed could be achieved over conventional bits of the time (Ref 19). For one particular set of tests in 1973, abrasive drilling rates for 8% abrasive concentration were 27 ft/hr at 5000 psi and 48 ft/hr at 10,000 psi. Using conventional bits, rates of 3 ft/hr using mud and 7 ft/hr using water, with 18,000 lbm on a 6½" bit were achieved (Ref 20).

The Gulf project finally terminated in 1975 for several reasons (Ref 20). Firstly, the abrasive caused severe equipment erosion, which made the system uneconomic. In particular, sufficiently reliable swivels and pumps could not be developed. Note that the abrasive slurry was pumped directly through the pumps. Secondly, a considerable cost was involved in reconditioning the return mud, the high viscosity required to maintain the steel shot in suspension making the removal of the shot for reuse very difficult. Finally, they showed that abrasive jet drilling would be economically viable only in the deeper wells, where conventional drilling rates were low.

3.1.2 The Flowdril System

Another major study of jet assisted drilling was conducted by FlowDril, a subsidiary of Flow Industries Inc. which started in 1985. They developed a system in which the drill string consisted of concentric drill pipe. The inner pipe conveyed small volumes of high pressure mud (30-40 gpm at up to 30,000 psi) to be ejected through nozzles in the mechanical cutter directly onto the rock face (Ref 21 and 22). The outer annulus carried conventional low pressure mud for hole cleaning.

To minimise component wear no abrasives were added to the mud Consequently very high pressures were required to ensure the rock was cut (compare 10,000 psi for abrasive jetting and 30,000 psi for non-abrasive jetting). These high pressures increase the expense and complication of the equipment and add further to the worries of rig safety (typically 5,000 psi is the maximum pressure encountered on an oil rig, although the equipment is rated to much higher pressures).

Although cost savings have been predicted due to the increased drilling rate, the reduction in bit wear and the consequent time savings in not having to trip so often to change bits (tripping out and back into hole can take as long as 12 hours), there are several major disadvantages with the FlowDril system:

- The concentric drill tubing is considerably more expensive to produce, and the tube sections are more difficult to connect on the rig.

- Drill pipe sizes were limited to 4½" to 5" in diameter, restricting the maximum depth of operation to 12500 ft - 15000 ft.

- The concentric pipe arrangement made fishing operations more difficult.

No further information has been obtained on the FlowDril system, however with the reliability and speed of current PDC bits, it is difficult to believe that this new system or a similar abrasive system would be widely accepted.

3.1.3 Downhole Intensifier

Veenhuizen, also from FlowDril Corp (Ref 23), describes a downhole pump for jet-assisted drilling. This is run on conventional drill pipe and sized for $8\frac{3}{4}$ " diameter holes. A double-ended intensifier pump has been developed capable of achieving 30,000 psi (2000 bar) at an output flow rate of 23 gpm (90 l/min). The intensifier is driven by the conventional mud stream, operating at 2000 psi higher than normal, and at flow rates of 320 gpm (1200 l/min). The small portion of high pressure fluid is directed to a nozzle in a specially modified drill bit.

Using this pump, rates of progress of between 1.3 and 2.5 times conventional rates have been claimed. This project was still in the development stage at the time when the article was written. However, to be successful the pump had to be shown to be reliable enough to be economic.

To operate such a pump on an abrasive fluid is unlikely to be economically feasible because of reliability problems. An abrasive free fluid supply would be needed to drive the intensifier. If the design had to be scaled down for use with coiled tubing, the final volume of high pressure abrasive fluid would be too small for effective cutting, unless a very small nozzle (< 0.4 mm in diameter) is used. Such a nozzle would be very susceptible to blockages and very accurate stand-off control would be needed.

3.1.4 The Rogaland Research DIAJET Drill

Rogaland Research conducted some small experimental trials on abrasive water jet assisted drilling systems (Ref 24). The abrasive slurry is generated by a 350 bar DIAJET system, removing the problem of having to put the abrasive slurry directly into the pumps, and fed through a ½" high pressure hose, installed inside the drill pipe, to 6 nozzles of 1.0 mm and 2.0 mm diameter. The nozzles are symmetrically installed in a nozzle holder mounted in the face of the drill bit (Figure 3.1.1).

As the mechanical cutting head is rotated, the flexible hose circles inside the drill pipe and the bearings between the nozzle holder and the bit ensure that very low torque is transferred to the hose due to the rotation. Consequently, no swivel arrangement is required.

The tests showed that the cutting is mainly due to the abrasive jets and that the mechanical cutter removes only remnants of concrete. Based on the energy required to produce holes in concrete samples, the researchers predicted that in hard rocks, with abrasive jet assistance, a 300% improvement in drilling rate could be achieved. Skaugen (Ref 25) gives empirical equations for the depth of cut and drilling rate that should be obtained with an abrasive jet drilling system.

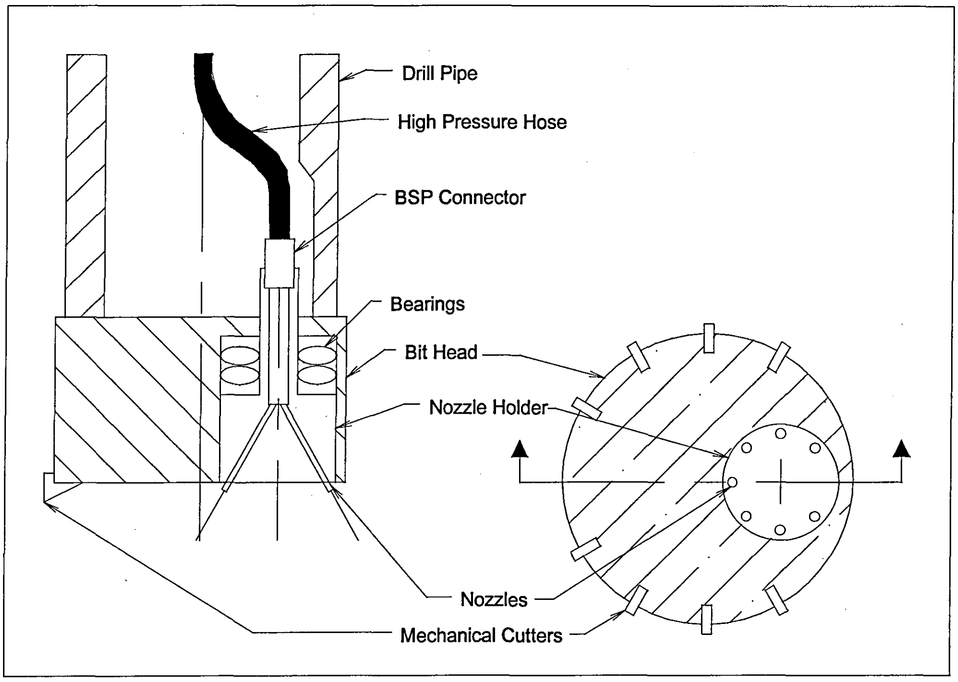


FIGURE 3.1.1 THE ROGALAND ABRASIVE DRILLING HEAD

The paper briefly discusses how a practical system could be developed. They recommend that the reliability of the DIAJET system would have to be improved, in order to ensure it could operate continuously for many hours, and that great care is needed to ensure that oversize particles do not enter the system. For their experiments, a high pressure filter on the slurry output side of the DIAJET is used. The greatest problems are envisaged to be in maintaining the abrasive in suspension, and ensuring that the abrasive and cuttings are removed from the well. (Note that a larger volume of abrasive would be used than the volume of cuttings produced.) Large quantities of abrasive would be required to drill a well and therefore to reduce costs an abrasive reclaiming system would be needed.

3.2 Directional Drilling

3.2.1 Conventional Directional Drilling

The above section covers examples of developments for conventional straight hole drilling. A major new area of development is the ability to conduct directional drilling, in which the drilling head is steered to the oil reservoir. The main advantages of directional drilling are that (Ref 26):

1) the well can be drilled along the length of a reservoir, significantly increasing the drainage area and consequently the production.

2) with accurately placed wells, production can be achieved at lower drawdown pressures (the difference in pressure between the reservoir and the wellbore), reducing sand production and the possibility of producing water or gas.

3) existing, uneconomical wells can be made profitable by reentering the well and drilling radial drain holes into the reservoir.

On average, a directionally drilled well is estimated to cost twice as much as a vertical well but 3 to 4 times the volume of oil can be recovered (Ref 27). However, greater risk is associated with horizontal wells and it is extremely important that the drilling head can be accurately controlled and directed to stay in the middle of the reservoir.

Successful and economic horizontal drilling has mainly been achieved with the development of two downhole tools. The first is a Measurement While Drilling Tool or Geosteering Tool, which consists of an array of sensors which measure the position of the bit and the formation rock characteristics. This information is then passed to the surface either by electric wireline (an electrical cable installed inside the drill pipe) or by mud pulses (Ref 28). The second tool is a downhole mud motor (Ref 29), usually a positive displacement motor, which generates rotary motion to drive the cutting bit by the pressure of the mud passing inside the drill pipe.

3.2.2 Coiled Tubing Drilling

See page 47, Section 4.7 for a description of Coiled Tubing.

Coiled tubing is gradually being used to drill new, shallow wells, for sidetracking, or deepening conventionally drilled wells and for reentering a well through the production tubing (Ref 30).

Coiled tubing has many advantages over conventional, jointed drill pipe, including the following:

a) coiled tubing can be deployed and retrieved more quickly, 100 ft/min compared to 30 ft/min, without the need for making up joints, which in the North Sea can take 15 minutes (Ref 31).

b) coiled tubing can be used in a live well, enabling new well bores to be drilled from existing, flowing wells. With conventional rotary drilling, the well would have to be killed, by adding a weighted fluid, to prevent the possibility of a blowout. This weighted fluid is often forced into the formation, which causes considerable damage. Coiled tubing drilling can be undertaken underbalanced, reducing the tendency for the fluid to pass into the formation. Conventional underbalanced drilling is possible, but it has proven safer and more efficient with coiled tubing (Ref 30).

c) less equipment is required with a coiled tubing rig. This makes it far easier to transport and means the site footprint is considerably smaller.

Coiled tubing cannot be rotated from the surface. Consequently, the drill bit is powered by a mud motor. Directional control is achieved using a bent sub, which can be altered to give the desired angle of deflection (Ref 32) and the weight on the bit is maintained by pushing the coiled tubing into the well.

3.2.3 Buckling Calculations

For both methods of directional drilling, computer calculations are conducted to predict the critical load that can be applied to the drill pipe or coiled tubing. Above the critical load, buckling occurs which results in a sudden increase in the force required to move the tubing. This force increases until a point is reached where the tubing "locks" (Ref 33). Then it can be pushed no further into the well.

3.2.4 Abrasive Directional Drilling - The URRS and QRS Systems

Petrolphysics developed two radial drilling systems which use high pressure water jets to cut radial boreholes into the formation from the vertical well. These systems were developed to provide a simple method of accessing more of a reservoir from an existing well and to access thin isolated reservoirs which were missed when the initial well was drilled.

The first system is the ultrashort radius radial system (URRS) (Ref 34) which has a turn radius from vertical to horizontal of about 1 ft (0.3 m), compared to about 30 ft (1 m) for short radius drilling, and places 100 ft (30 m) to 200 ft (60 m) radial boreholes into the formation. At the required depth in the well an underreamed cavity is cut, into which an ultrashort radius whipstock is erected (Figure 3.2.2) (Ref 35). A 100 ft to 200 ft length of 1¼" coiled tubing is lowered on wireline into the casing and into the top of the whipstock. Connected to the end of the coiled tubing is a conical jet nozzle with water jet side thrusters, activated by solenoid valves, which enable the nozzle direction to be controlled.

Once the system is pressurised to the working pressure of 5000 psi to 10,000 psi (345 to 690 bar) (typical volume of 160 gpm, 0.36 m^3/s), the side thrusters provide the force to move the coiled tubing around the whipstock and out horizontally into the formation, the conical water jet cutting a borehole of 2" to 4" in diameter. In consolidated formations the radial borehole can then be cased using the coiled tubing. Electrochemical cutting methods are used to cut and slot the tubing and then it can be gravel packed to prevent sand production.

A system of such boreholes can be placed into the formation. They have the advantage of penetrating beyond any wellbore formation damage, do not damage the formation themselves and consequently enable an increase in oil production. Reference 36 quotes an average improvement in production of between 200% and 400%.

The second system, which is less developed, is the Quick Radial System (QRS) which also drills multiple radials using water jet drilling and coiled tubing, but uses a larger radius of turn and does not require the milling out of a section of casing or the underreaming of a cavity in the formation to work. Instead, an abrasive jet at between 3000 psi and 5000 psi (200 to 345 bar) is used to cut through the metal casing and enter the formation. Plain water is then used to cut the rest of the borehole.

From the information obtained, the URRS has only been used in wells up to 1500 ft (500 m) deep and oil companies have indicated that the operating pressures are too low for effective cutting. Petrolphysics have tried to minimise the use of abrasives because, on occasions, they had experienced plugging of the nozzles and instead have developed cutting fluids with polymer additives to maximise their cutting ability. They have also had problems with buckling of the coiled tubing as it is pushed along the well bore and have had to use their side jets to help to keep the tubing in tension. (Ref 37).

A similar system to the URRS is currently being developed by the US Navy to drill from land out to sea to provide conduit protection for offshore cable installations (Ref 38). This system initially used coiled tubing to convey the cutting fluid but for extended reach drilling 4½" drill string was needed to reduce friction and buckling. Operating pressures of between 10,000 and 15,000 psi are being used.

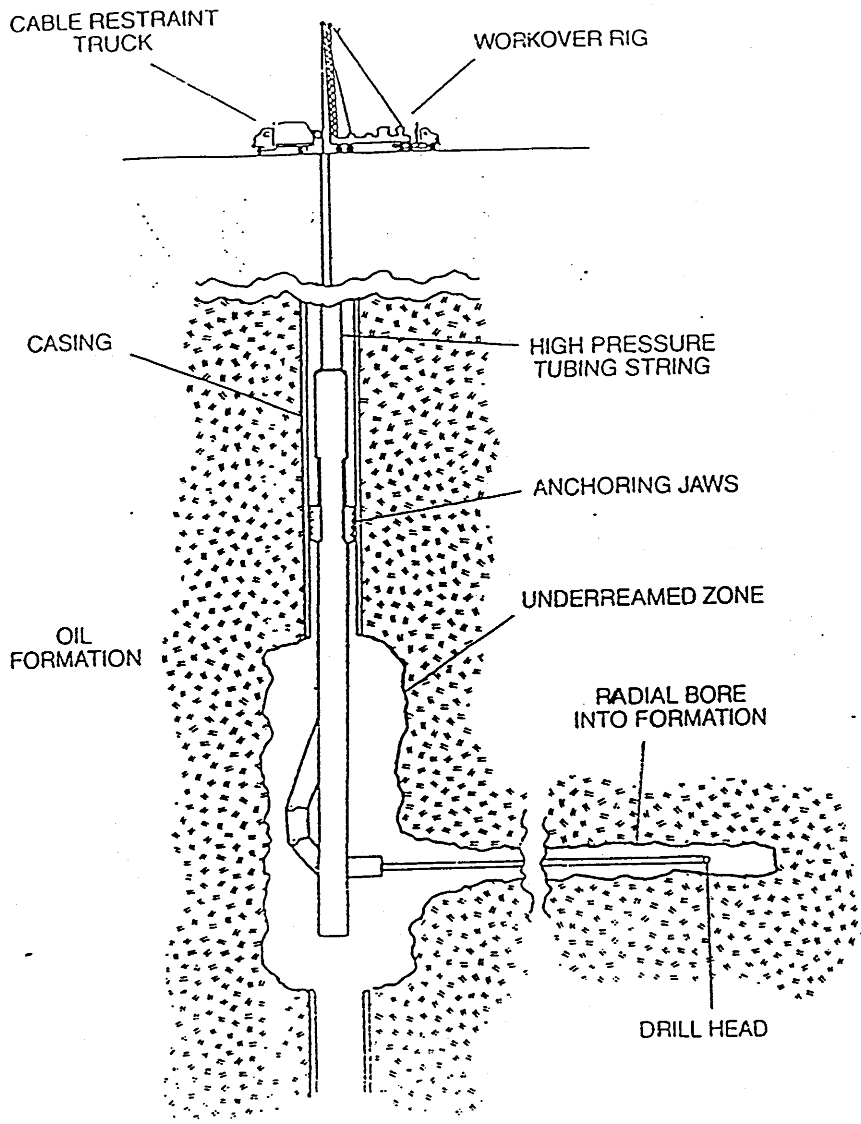


FIGURE 3.2.2 THE URRS SYSTEM BY PETROLPHYSICS

3.3 Perforating and Fracture Initiation

3.3.1 Completing the Well

After a section of well has been drilled, steel tubing, called casing, is cemented in the well. This ensures that the well bore cannot collapse and protects the formation from further damage and contamination by the drilling fluids.

This casing and cementing operation is repeated at required intervals until the oil bearing formation is reached. The well can then be completed by one of the several configurations described below in order to allow oil to enter the production tubing (Ref 39).

1) Openhole completion, in which the production formation is not isolated by the casing, which extends to the top of the formation only (Figure 3.3.1a). This is a cheap solution and is only used for competent, permeable formations (Ref 40).

2) Slotted liner completion, in which steel tubing containing slots is hung from the casing and is not cemented in place (Figure 3.3.1b). These are used in permeable formations where further work to stimulate the well is unnecessary. They are also very useful in long horizontal sections, which can be expensive and difficult to complete by other means (Ref 41). If the slots are made small enough (<0.2mm wide) this method can also be advantageous for sand control. Grains of sand tend to bridge the slot which prevents other grains from entering the well with the oil.

3) Cased and perforated completion, which involves cementing casing in the oil bearing formation and then making holes or perforations through the casing and into the formation to enable the reservoir fluids to enter the well. This is the most common and most expensive method of completion but provides the flexibility of being able to select the sections to be perforated (Figure 3.3.1c and d).

An abrasive water jet cutting system can potentially be used in cases (2) and (3). The very small kerf widths that can be achieved using an abrasive water jet cutting system mean that slotted liners could be produced. However, for this discussion, only downhole cutting applications will be reviewed.

3.3.2 Perforating with Explosive Charges

Casing is generally perforated using shaped explosive charges, similar to bullets, which are carried on a perforating gun. Over 1300 perforations can be placed at the touch of a button, each one generally being greater than 8" long and 0.25" in diameter.

Apart from the obvious dangers of using explosives in a highly unstable environment and the difficulties associated with transporting and using them in some of the most volatile parts of the world, explosives used for perforating have the disadvantage of damaging the formation which results in a reduction in productivity. This is described in detail in many papers, for example References 42 and 43.

In difficult to produce formations, perforating alone is insufficient to ensure good productivity. After perforating, the well is therefore pressurised to force open cracks in the formation generated by the explosive charges. This is called hydraulic fracturing. If required, proppant particles, such as special sand, are then pumped into the cracks to keep them open.

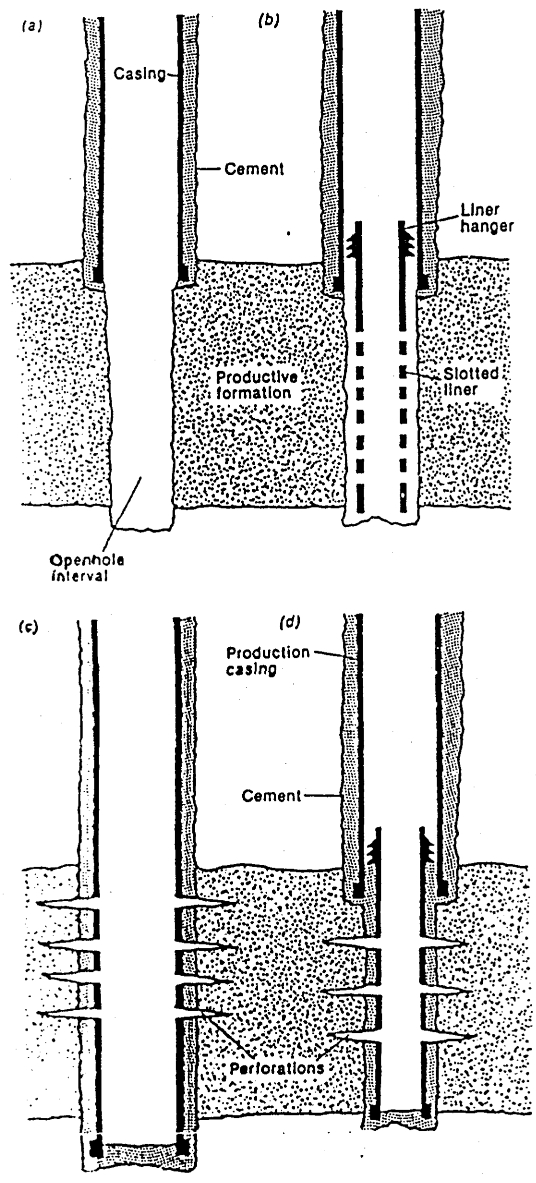


FIGURE 3.3.1 BASIC COMPLETION CONFIGURATIONS.
a) openhole, b) slotted liner, c) and d) cased and completed completions.

3.3.3 Abrasive Water Jet Perforating Tools

The potential attraction of an abrasive water jet perforating tool is the possibility of producing perforations which are damage free. Considering that permeability can be reduced by more than 20% by using explosive charges, producing perforations of similar permeability to the virgin formation would translate to a considerable increase in production rate.

Abrasive water jet perforating and fracture initiation has been investigated by many researchers, including Pittman (Ref 44), Pekarek (Ref 45), Ousterhout (Ref 16) and Surjaatmadja (Ref 46).

Their perforating experiments generally consisted of directing a stationary or slow moving abrasive water jet at a target consisting of a steel plate cemented to a block of berea sandstone. (Note that berea sandstone is the recommended rock on which to conduct perforation tests, Ref 47.) The abrasive water jet was generated at pressures varying from 3000 psi to 5000 psi, using nozzle diameters from 4.8 mm to 6.4 mm, and were conducted submerged and under pressure to simulate perforating at depths of up to 2000 ft (back pressure of 66 bar).

Their findings can be summarised as follows:

1) Typical depths of cut were between 5" and 7" for 15-25 minutes of jetting time.

2) The depth of cut was generally found to decrease as the back pressure increased up to 150 psi (10 bar), above which further increases in back pressure had little affect. This confirms the findings in Section 2.6, although no general relationship between back pressure and depth of cut is given.

3) Acceptable nozzle life could be achieved by using tungsten carbide nozzles.

4) Abrasive jet perforating is claimed by several of the authors not to compact the formation, implying that the permeability is not changed. No information is given on how this was determined.

5) The perforations or notches produced in the formation were found to be ideal fracture initiation sites, producing planes of weakness in the formation which could be more easily opened by hydraulic fracturing than normally produced perforations.

   Although the time taken to cut these notches is far longer than by using conventional techniques, for certain low producing wells a few perforations generated using this method may increase production enough that the jetting time is not critical.

Surjaatmadja describes a special fan type jetting tool which can cut almost a full circular slot in the rock. This has been experimentally shown to be ideal for producing a single perpendicular fracture. No information is given on the pressures and nozzle sizes that were used.

6) In practise, abrasive jet perforating has generally been conducted in hard formations in which "regular perforating methods failed" and also used to selectively perforate thin oil sections in gas-oil wells in which conventional methods produced practically all gas.

The Lance Tool and the Punch-Jet Tool

Peters (Ref 48) describes two liquid jet cutting tools for producing large perforations or tunnels into the reservoir formation; the lance tool and the punch-jet tool. Each tool utilises a metal punch to puncture the casing and then high pressure liquid (up to 10,000 psi) is used to jet away a hole into the formation with no compaction. On one of the tools, the nozzle is connected to a flexible hose so that it can be directed through the punch and into the formation. Perforations up to 10 ft long can then be generated by maintaining a stand-off distance of less than 2 inches.

He claims that the tool is ideal for three situations (Ref 49):

'1) Enhancing the flow in injector wells, where high conductivity with the reservoir is needed.

2) Heavy oil production from soft formations, where maximum formation exposure must be obtained through a heavily invaded zone around the borehole.

3) Zones where cement, paraffin or asphaltene deposits are heavy.'

However, Crook (Ref 50) found from discussions with oil service companies that the penetrator tools were not that effective, the operating pressure being too low.

Feasibility Cutting Trials at DIAJET Limited

I conducted some feasibility perforating trials on sandstone blocks, with a 10 mm thick steel plate clamped to them. The blocks were submerged in 300 mm of water and cut using a 1 mm diameter nozzle, with different pressure drops, stand-offs and traverse speeds.

The trials showed that a 90 mm deep slot could be cut in the sandstone block with a 690 bar nozzle pressure drop, a traverse speed of 100 mm/min and a stand-off of 10 mm. It is important to note that if this test was repeated at a back pressure of 350 bar with the same pressure drop across the nozzle, a depth of cut of between 40 mm and 60 mm would be expected.

These tests confirmed the findings in the literature review: **an abrasive jet perforating tool is too slow for practical use**. The depths of the slots cut in these tests are also too shallow to be effective perforations, if it is assumed that a minimum perforation depth of 200 mm is required. However, as a crack inducing tool, these shallow slots may be ideal.

Microscopic analysis of the sandstone surface cut by the abrasive jet did not reveal any abrasive particles embedded in the voids. There was also no indication that the voids had been filled by crushed particles of sandstone. Although the permeability of the sandstone could not be measured, there was no evidence of surface damage and therefore the conclusion that abrasive water jet cutting does not crush the surrounding formation is probably correct.

For some difficult to produce formations, an abrasive water jet fracture initiation tool may therefore be advantageous. However, discussions with potential users revealed concerns about how such a tool could be accurately moved in the well and whether the jet could be quickly turned on and off between slots.

3.4 Descaling of Casing

Some wells experience scale build up on the inside of the production tubing during oil production, which decreases the bore and causes the oil flow rate to drop. Softer, carbonate scales can be dissolved using acid but this requires production to be stopped for anything from 24 to 36 hours while the acid is circulated (Ref 37). Mechanical downhole reamers can also be used, although their success is variable, particularly for the harder scales such as barium sulphate. The only alternative then is to remove the casing and clean the tubulars on the surface, special disposal facilities being required because the scale is often radioactive.

High pressure water jetting tools mounted on coiled tubing have been used to clean tubulars, for example the OTIS Indexing Jet Cleaning Tool (OTIS is a Halliburton Company) (Ref 51). However, their performance on very hard scale and at significant depths, where cavitation is negligible, is poor.

Some investigations have been conducted with abrasive jets for the removal of the hardest scales but success was limited. Difficulties were encountered in trying to produce a multiple jet array, in order to cover the whole circumference of the tubing, which could effectively operate submerged and remove scale at an acceptable rate. Also, the pressures required to reliably remove any type of scale were high enough to damage the steel casing, which is obviously unacceptable.

3.5 Cutting and Decommissioning

Section 1.2 mentions some of the wide ranging industrial applications for abrasive water jet cutting tools since their commercial development around 20 years ago. Of obvious interest to the oil industry was the cold cutting ability of abrasive water jets and their suitability for remote operations in potentially explosive situations.

3.5.1 Abrasive Jets for Blow-Out Control Operations

High pressure abrasive jet cutting tools were prominent in Kuwait in 1991 when burning wells had to be extinguished and brought under control. They were used to cut away damaged casing and well head equipment so that new blow-out preventers could be mounted.

References 52 and 53 describe two systems specifically designed for blow-out control operations, one made by Harben systems, the Hytorc-Harben Jet Edge, and the other by Halliburton Services. These systems operate at over 10,000 psi and are designed such that the abrasive is added after the pump. Reliable and rugged linear actuators were a key to the success of the tools.

Examples of cutting performance include the cutting of 20", $13\frac{3}{8}$", $9\frac{5}{8}$" and 7" casing spool with $3\frac{1}{2}$" tubing in 1 to 2 hours and the cutting of a 24"OD by 13" ID flange (equivalent to $5\frac{1}{2}$" wall thickness) in 119 minutes, two passes being used to ensure a complete cut. The main problem with casing string cutting was encountered if the strings were not completely cemented together. Any voids in the cement resulted in the jet's energy being dissipated in the void.

3.5.2 Platform Removal and Well Abandonment

Abrasive water jet cutting systems have also been used in the decommissioning of wells and oil platforms. Stolt Comex Seaway (Ref 54) used a DIAJET system to cut up to 140 mm thick steel around the base of the Furmar Buoy, 80 m underwater. Cutting speeds were set deliberately low so that a complete cut was ensured; no reliable method of monitoring a cut has been found when access is available to one side of the target only. As usual with abrasive jet cutting, the greatest problems were associated with designing a suitable manipulator for the nozzle. For this particular task, the only alternative to abrasive jet cutting would have been to use explosives to cut the metal but this involves added danger and is less effective in very thick sections.

When a well is ready for abandonment, the Xmas tree is removed for possible reuse. Usually the casing string is cut internally, approximately 10 ft below the surface, or mud line for a subsea well. Hotforge, part of the Red Baron Group, have used a DIAJET and their own manipulator for internally cutting casing strings, the slurry being supplied by a $\frac{3}{4}$" umbilical rated at 5000 psi. Unfortunately, as described above, they have experienced difficulties cutting the complete casing string (typical arrangement given below) because of voids in the cement between the casing.

Typical casing string arrangement:

$9\frac{5}{8}$" OD	0.312" - 0.75" wall
$13\frac{3}{8}$" OD	0.33"-0.514"wall
20" OD	0.5"-0.635" wall
30" OD	1"-1.5" wall

The environmental acceptability of abrasive jet cutting is of particular importance in the abandonment of water wells. There is concern that some wells have not been isolated and protected adequately (Ref 55), leaving the possibility of the contamination of water sources. However, jetting tools have been used to remove existing casing to enable cement to be pumped into the well to isolate the water source.

3.5.3 Explosives

The main method of cutting the piles of offshore structures is to use explosives. Explosives can be positioned relatively quickly, but can be subject to extensive delays to prevent harm to marine mammals and turtles (Ref 56). The area around the structure must be monitored for at least 48 hours, with any sea life being safely removed or allowed to leave, before the explosives can be detonated. In US Coastal waters, explosions are also limited to daylight hours only.

Even so, nearly 70% of platforms dealt with in the USA have been removed with explosives (Ref 57). This article presented a cost comparison table from a National Research Council Report on pile cutting methods. They concluded that:

"Explosives	very reliable, high experience level, excellent safety record, lowest exposure time of all methods. Total costs $1030;
Mechanical cutting	prone to problems, very labour intensive, safety concerns. Total costs $1497;
Abrasive cutting	not very reliable to date, still in test phase. Total costs $1270."

The total cost is based on the various procedures involved in the removal, including decommissioning, derrick barge removal, direct severing and positioning.

Explosive technology is progressing rapidly. By using specially shaped charges, better cuts can be obtained with less impact on the surroundings. (Ref 58)

Overall, considerable development work is required on abrasive water jet cutting techniques in order to compete with explosives.

3.6 Summary of Using Abrasive Water Jets Downhole

The above review identifies some of the most important factors which must be considered if abrasive water jets are to be used to cut a window in oil well casing. They are:

1) The abrasive particle size must be carefully controlled to ensure there is no danger of nozzle blockage.

2) The cutting fluid must be selected to ensure the abrasive does not settle if pumping is stopped. The settling of the abrasive could complicate removal of the jetting tool and even endanger the well if it cannot be removed. Some oil companies seemed particularly wary about using a tool that, initially at least, puts more debris into the well.

3) The cuttings must be consistently small enough to be reliably removed from the well.

4) For efficient cutting, the highest nozzle pressure should be used. This is limited by the allowable surface pressure, the pressure limits of the coiled tubing or drill pipe, and the friction losses which depend on the depth of operation.

5) The quantity of abrasives, and consequently cutting fluid, that would be required. For long cutting operations, the cutting fluid will have to be reconditioned and reused (all of the abrasive must be removed first). Recyclable abrasives might have to be considered, depending on the economics.

6) The life of critical components, such as nozzles, is important. A considerable amount of time, and therefore money, is wasted if tools have to be tripped in and out of a well to replace the worn components.

These factors are considered for an abrasive water jet window cutting system in detail in Chapter 5. The next chapter discusses the techniques used for cutting a window and identifies a potential market for an abrasive water jet window cutting tool.

4.0 WINDOW CUTTING

4.1 Introduction

Window cutting is the cutting of a hole in the side of the casing inside the well so that a drill string can then be guided through it. This is necessary if directional or horizontal wells need to be drilled to improve the production of an existing well, or if a sidetrack is required to avoid objects in the well which cannot be removed, such as stuck pipe.

Multi-lateral completions are becoming a popular method of maximising production from a well. It was very noticeable at the Offshore Europe Exhibition which I visited in Aberdeen in October in 1995 that all of the major oil service companies were advertising their new multi-lateral techniques. Where these were being applied to existing wells, windows need to be cut in the casing. The ability to reliably cut a window has therefore become of increasing importance.

This chapter reviews the different window cutting techniques, including recent attempts at using abrasive water jets, and identifies a specific market for a new abrasive water jet window cutting tool.

4.2 Conventional Window Cutting Techniques

4.2.1 Introduction

Window cutting techniques either use drill pipe or coiled tubing to convey the cutting tools. Each method has its advantages and disadvantages.

Drill pipe requires the use of a work-over rig, which is expensive to hire and can be difficult to transport to isolated regions. For sidetrack operations the production tubing often has to be removed. However, large torques can be applied and the weight on the bit can be controlled, although this becomes more difficult in deviated and slim-hole wells.

Coiled tubing offers the possibility of entering through the production tubing. No work-over rig is required and the cost of the operation is greatly reduced as hire charges are much lower and coiled tubing units are much easier to transport. If the production tubing has to be removed, a coiled tubing unit can still be used for cutting the window as money is saved by hiring the work-over rig for as short a time as possible. Further advantages can then be achieved by directionally drilling with coiled tubing rather than drill pipe. The main problem with coiled tubing is applying and controlling sufficient weight and torque on the bit to cut through the metal casing.

The methods used for cutting a window using drill pipe or coiled tubing deployed tools will be briefly explained in the next section.

New wells are being designed with sidetracking operations in mind (Ref 59). In these cases a special type of casing is used which already has a window cut into it. The window is covered with a composite, such as fibreglass, which is much easier to cut than steel. By using wireline survey instruments, the windowed casing can be lowered into the well and orientated as required. Modified conventional window cutting techniques, which are described below, are then used to cut through the window and drill the new lateral well.

To maintain the pressure integrity of the casing internal pressure sleeves can be installed. Then, once the window is ready to be cut, the sleeve is moved to expose the composite cover.

4.2.2 Cutting the Window

Different types of cutting tools, called mills, are used to ensure that the correct size of window is cut with no jagged edges or burrs being left on the casing. Each type of mill has a specific task in the window cutting process.

The basic principle of cutting the window is to direct or deflect the cutting tools, using the different methods described below, to the desired point on the casing wall where the window is required. Cutting can then begin.

4.2.3 Window Cutting with Drill Pipe Conveyed Tools

Brock (Ref 60) describes cutting a window using drill pipe to convey the cutting tools.

Below the required position for the window, the well is cemented across. A packer is then lowered into the well on electric wireline and activated to clamp against the casing wall. This packer provides a firm base for the whipstock. The drill pipe, mills and whipstock are then run into the well (Figure 4.2.1). An anchor in the base of the whipstock is aligned with a guide key in the packer. When the anchor has engaged in the key, the drill pipe is raised and a shear bolt, connecting the drill pipe to the whipstock, is sheared, leaving the whipstock fixed firmly in the packer.

The whipstock is a tool that has one face angled, for example at 30°, so that as the drill pipe is pushed downwards, it is deflected down the face towards the casing wall. By measuring the direction of the guide key in the packer, the anchor can be welded to the base of the whipstock in the correct position relative to the angled face. The drill pipe will then be deflected by the whipstock in the required direction.

Drilling mud is then pumped into the well and the drill pipe is rotated to start milling. Careful control of the rate of progress, the weight on the bit and the applied torque ensures that a successful window is cut. Several different mills are required to cut, elongate and complete a window and to reduce the number of trips, two or three mills can be run on the drill pipe together. Note that correct alignment of the milling axis has been found to be very important if the whipstock is not to be cut (Ref 61).

Baldauf (Ref 62) states that for a 5.91" (150 mm) diameter mill, milling a $7\frac{5}{8}$" Q125 liner a milling rate of 0.25 m/hr is reasonable.

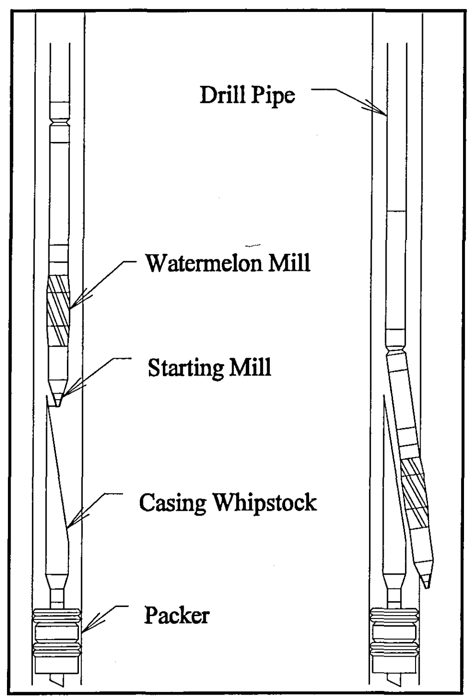


FIGURE 4.2.1 WINDOW CUTTING WITH DRILL PIPE CONVEYED TOOLS

4.2.4 Window Cutting Using Coiled Tubing Deployed Tools

Coiled tubing window cutting was developed to enable windows to be cut below the production tubing without having to remove the production tubing. This is called Through-tubing Window Cutting. If coiled tubing is used to convey the milling tools then a downhole motor is required to provide the rotation necessary to drive the cutting tool. However, because of the limited torque produced by the motor, only one or possibly two mills can be powered and consequently separate trips are required.

Careful control of the weight applied to the mill bit is very important. Enough force must be applied to enable the bit to "bite" into the metal and form a lip in the casing wall. Too low a force and the bit will skate over the surface, too high and the bit will just slide down the casing. At the end of cutting operations it is essential that a smooth exit has been cut, as there is always the chance that tools could become stuck on any ledges or burrs left on the casing or cement.

To maintain directional control at the start of milling, time drilling is used, where the depth is increased in small increments over a set period, for example 0.5 to 1 ft per hr. Accurate control of the coiled tubing is very difficult at such slow speeds and is best achieved by reeling out the tubing in 0.1 ft intervals and then applying the brake (Ref 63). The milling bottom hole assembly can be as small as 2.875" OD.

There are three methods of deflecting the coiled tubing to the casing wall. These are described by Leising (Ref 63) and Faure (Ref 64). They are:

i) Cement Sidetracking
ii) Whipstock in Cement
iii) Through-tubing Whipstock.

i) Cement Sidetracking

Cement sidetracking involves cementing across the casing and directionally drilling with a bent housing motor through the cement to the casing wall (Figure 4.2.2). Generally, this is a very simple and cheap method of cutting a window and, if required, the cement plug can be removed afterwards to allow access to the rest of the well.

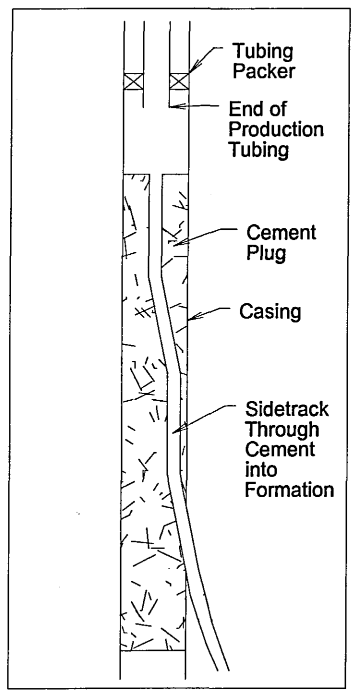


FIGURE 4.2.2 CEMENT SIDE TRACK

The lengths of the windows cut with this technique have, so far, been short (in the order of 1 m long instead of the required 2 - 2.5 m) which could hinder reentry. If too shallow a cutting angle is attempted, in order to cut a longer window, the drill bit slips off the casing and cuts down through the cement plug. The fragile cement plug is also susceptible to wear due to the continued entry and withdrawal of the coiled tubing, which increases the chance of tools becoming stuck downhole or deviating down through the cement instead of through the window.

ii) Whipstock in Cement

This is a development of cement sidetracking. Again the well is cemented across and then directionally drilled to the casing wall. Once at the casing wall, a straight section is drilled. A small diameter whipstock is then conveyed on the coiled tubing, through the production tubing and cemented section and anchored in the straight section at the correct orientation. Milling operations are then conducted as before, the whipstock providing more substantial support for the milling tools than the cement alone (Figure 4.2.3).

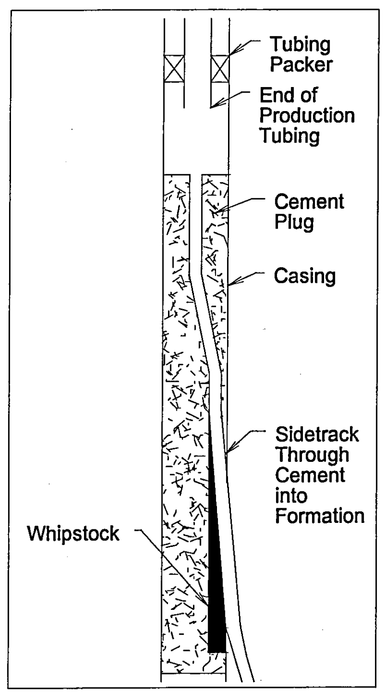


FIGURE 4.2.3 WHIPSTOCK IN CEMENT

iii) Through-tubing Whipstock

Through-tubing whipstock uses a whipstock with a specially designed anchor that can pass through the small bore production tubing and then expand to span the larger bore casing (Figure 4.2.4). Generally, whipstock and anchor orientation is achieved by gravity, the heavier side of the whipstock falling, by means of a swivel, to the lower side of the casing. This results in all exits being near the high side of the casing. Once the anchor and whipstock are correctly set, milling can be started as usual.

This method is more expensive than the others, but is ideal where production must be maintained. Once operations through the window have been completed, some types of whipstock can be retrieved and reused. Obviously, if no production tubing is present, simpler full size whipstocks can be conveyed on coiled tubing.

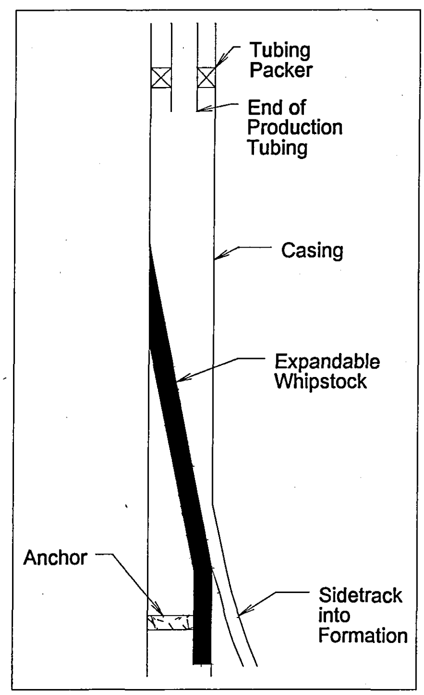


FIGURE 4.2.4 THROUGH -TUBING WHIPSTOCK

The use of coiled tubing to convey window cutting tools is still relatively new and their performance appears to vary considerably, although there seems to be a general improvement as operators gain experience of the techniques. Eide (Ref 61) quotes that a $5\frac{7}{8}$" (150 mm) window was cut in $6\frac{5}{8}$" and $9\frac{5}{8}$" casing in a total milling time of 12 hours, less than 45 hours being required for the whole window cutting operation. Hightower (Ref 65) quotes that a window 15 ft (4572 mm) long was cut in 5.5" OD J55 casing in 12 hours (381 mm/hr).

More recently, Townsend (Ref 66) quotes a series of tests that were conducted to mill a window through 7" and $9\frac{5}{8}$" casing using three different 3.75" whipstock and coiled tubing assemblies. All equipment had to pass through 3.75" restrictions in the 4½" completion tubing string. Typically, 12 ft long windows were cut, of which approximately 4 ft was in the $9\frac{5}{8}$" casing, in approximately 30 hours (average cutting rate of 0.4 ft/hr).

4.3 Window Cutting with an Abrasive Fluid Jet

4.3.1 Investigations by Rogaland Research

Rogaland Research have recently adapted their mechanically assisted abrasive drilling head (Figure 3.1.1) for through-tubing window cutting (Ref 67). The milling head is deployed on coiled tubing and is directed to the casing wall using the cement sidetracking principle described above. It contains 5 nozzles positioned on only one side of the holder, instead of being evenly distributed around the circumference. Consequently, an off-centre path is cut by the abrasive jets. The opposite side of the cavity to that cut by the abrasive jet is then widened to the correct diameter by the mechanical cutter.

This window cutting technique relies on the jets performing the initial cuts in the steel casing. As cutting progresses further milling is done by a combined mechanical and jet action. The asymmetric nozzle arrangement results in the lower side of the window being cut entirely by the mechanical cutter, so that a smooth sliding surface in the cement is formed to facilitate reentry through the window.

The advantage with this system is that it not only cuts the window but can also be used to directionally drill into the formation. The jet bit effectively steers itself, the conical shaped cavity produced by the jets acting as a guide for the drill bit. Also, because less metal is removed by mechanical means, the torque and the weight on bit needed is considerably reduced compared to conventional milling, simplifying control further. However, there is a danger that if the drilling rate is too high the mechanical cutter does not have the time to widen the slot. Consequently the bit would always be deflected into the cavity and a spiral path would be cut.

Only details of surface experiments have so far been published. A 1.4 m long window was successfully cut in 7" casing using 90 l/min of cutting fluid at 300 bar, 5% abrasive concentration and a rate of progress of between 100 - 150 mm/hr.

One problem that could be experienced with this system is that the intersecting jets might either generate some large cuttings, which would be difficult to remove from the well, or that some metal near the nozzle holder is left uncut. Due to the position of this uncut metal, it could prove difficult to remove mechanically.

4.3.2 Investigations by TIW

The Texas Iron Works Company (TIW) (Ref 68) hold a patent for a device which provides jets of abrasive fluid in a well (Figure 4.3.1). The device has been patented for use for counter-boring or under-reaming, window cutting in well liners and perforating them. It is the only device that I have found that uses just abrasive jets to cut the window.

The device potentially solves one of the main problems with any AWJC operation: the maintenance of a minimum stand-off distance between the nozzle and the target to be cut. This is achieved by having a telescopic nozzle which extends under pressure. The nozzle assembly is contained in a spherical body, similar to a ball valve. Then, when cutting has finished, if the nozzle meets a restriction as it is withdrawn from the well, the nozzle will rotate into the main body of the device. In this way it can be deployed through a no-go nipple.

No information has been obtained on whether this design has actually been tested. The sealing arrangement of the telescopic nozzle and spherical ball and seat are critical, especially as they have to seal in an abrasive environment. However, because each moving component only moves once during the full operating cycle inside the well, wear of the seals and the spherical ball may not be a problem. They can be checked and replaced if necessary after each deployment.

The patent describes a sequence of operation when the device is connected to a drill string. The assembly can then be rotated by a power swivel and lifted using a hydraulic servo compensator from the surface (these are not suitable for coiled tubing). Whether this method of manipulation could give the required accuracy to cut a window is doubtful.

The patent also gives an example of possible tool dimensions. Typically the device would be over 5" in diameter and would run a $\frac{3}{16}$" (4.7 mm) diameter nozzle.

4.4 Patent Search

An extensive patent search was conducted on manipulators for downhole abrasive water jet cutting and no relevant patents were found, except for the TIW patent described in Section 3.4.3. In particular, no patents were found for accurately controlling the motion of the cutting nozzles.

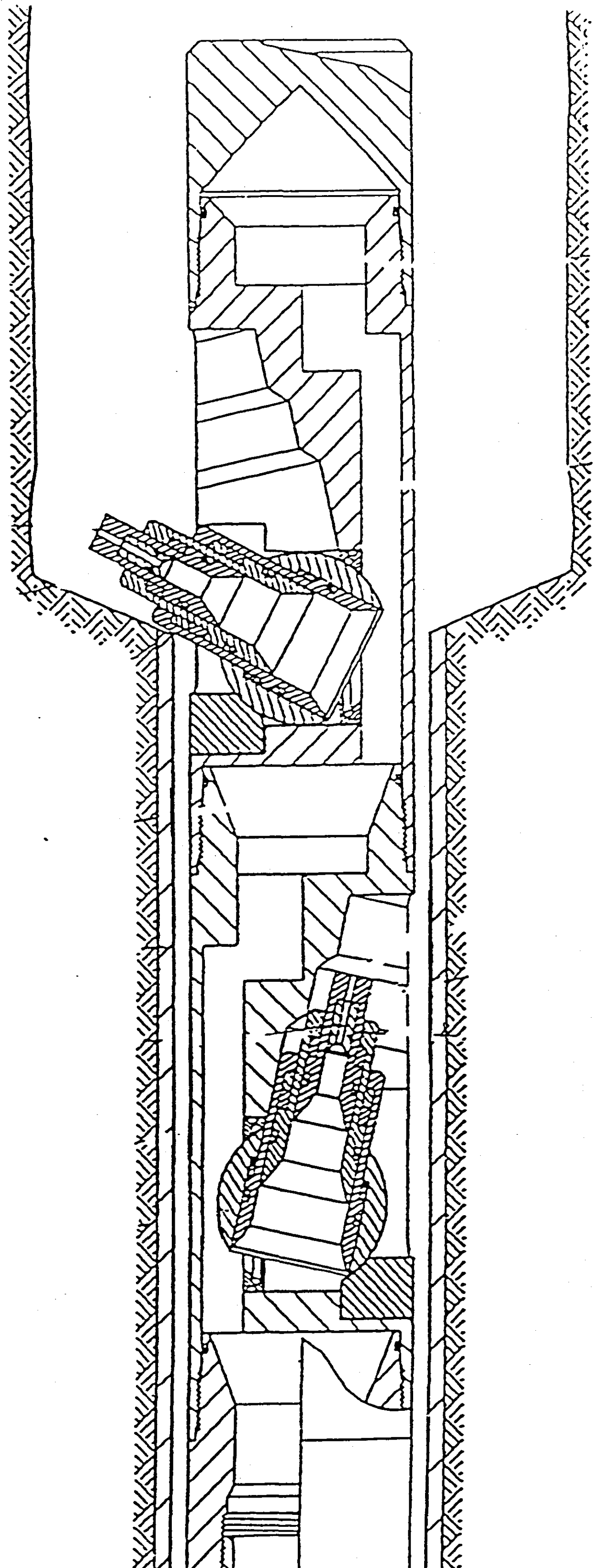


FIGURE 4.3.1 THE TIW ABRASIVE WATER JET CUTTING DEVICE

4.5 Potential Market for an Abrasive Water Jet Window Cutting Tool

The current low price of oil and the ever increasing costs involved in developing an oil field in the 1990's has forced oil companies to investigate new technologies which can either:

- economically increase the production of existing wells

or

- reduce the costs of developing new wells.

There is therefore a significant market for any tools which can assist in achieving either of these goals.

4.5.1 Oil Well Review

The distance an abrasive water jet cutting tool can be used from the surface will be limited by the pressure rating of the coiled tubing, as shown above. It is therefore important that the maximum operating distances that are likely to be encountered are determined.

Information on the wells drilled around the world and their general depths has been reviewed (Refs 69, 70 and 71), although from the statistics available only rough estimates of average well depths can be made. Unfortunately, no similar information has been found on the average length of wells. However, considering that horizontal technology is still relatively new, average well depths should give a good initial indication of what is required.

Overall, probably close to 70% of all wells drilled on-shore are less than 6000 ft (2000 m) deep. For example, 65% of wells in the USA are under 5000 ft. Although technology is forever progressing to enable deeper and deeper wells to be developed, perhaps only 10% of wells are deeper than 9000 ft. In countries such as Bolivia and Iraq, however, approximately 42% and 66% of wells are deeper than 9000 ft (2700 m) and wells over 17,000 ft (5000 m) deep have been drilled, for example in Venezuela.

Offshore, the average depth of wells is probably 8000 ft (2400 m). In the UK and Norwegian sectors of the North Sea, the average well depth is 8300 ft in an average depth of water of 300 ft (95 m), although only 25% are deeper than 9000 ft (2700 m). New wells planned for the West of Shetland will be approximately 10,000 ft (3000 m) deep, situated below 1000-2000 ft (300- 600 m) of water. The average depth in the USA is 9300 ft.

These results indicate the importance of knowing the operating limits of an abrasive water jet cutting tool so that the final cutting operations can be chosen to be compatible with the markets for which the tool is being aimed.

4.5.2 A Small Diameter, Through-Tubing Window Cutting System

My Supervisors obtained a survey (Ref 72) showing that in North America, Canada and Alaska there are 908,000 wells, the majority being land wells. The survey showed that:

51.6% have $2\frac{3}{8}$" production tubing,

41.3% have $2\frac{7}{8}$" production tubing,

6.3% have $3\frac{1}{2}$" production tubing,

0.7% have greater than $3\frac{1}{2}$" production tubing.

As US oil companies look to maximise production from reservoirs and even reopen abandoned wells (Ref 73), the potential market for a miniature through-tubing drilling and window cutting system is enormous.

Note that in many shallow cases, drilling a completely new well may be as cheap as reopening abandoned wells. Each well would have to be assessed on an individual basis.

Although 2" drilling equipment is currently available, no information has been obtained on similar window cutting equipment. The References given in Section 4.2 all describe operations through a minimum of 3.75" production tubing and Faure (Ref 64) quotes that in 1993 re-entry drilling systems were only available down to $3\frac{7}{8}$". Even the abrasive jet assisted milling tool being developed by Rogaland will only pass through a minimum $4\frac{1}{2}$" production tubing.

As explained in Section 4.2.2, controlling the weight on the mill bit is very important for conventional milling. For small diameter, 2" bits, control would become even more difficult. An abrasive water jet window cutting tool, which requires no weight to be applied, could be a far more effective and reliable alternative to developing smaller conventional milling tools.

If a suitable manipulator can be designed to accurately control the nozzles, it should be possible to cut a window of any size. With current milling tools, there is no method of guaranteeing the size of the window.

In order to compete with existing, although larger, coiled tubing deployed window cutting tools, an abrasive water jet window cutting system would need to be able to pass through a 2.205" No-Go Nipple installed in the production tubing, and cut a 6 ft to 7.5 ft (2 m - 2.5 m) long by approximately 3.75" (95 mm) wide window in 5" to 7" diameter casing. Typical casing wall thickness is 6 mm to 10.7 mm. The complete window cutting operation should be completed in a target time of 10 hours (cutting rate of 0.6 ft/hr).

A feasibility study for a through-tubing abrasive water jet window cutting system is presented in the next chapter.

To complete the review, possible methods of conveying the cutting fluid to the nozzle, suitable cutting fluids and the essential tools which must be deployed with coiled tubing must be considered.

4.6 Conveying the Cutting Fluid to the Nozzle and Downhole Communication

The OD of the window cutting tool must be less than 2". Coiled tubing is therefore the only current method of conveying the cutting fluid to the nozzle, drill pipe is too large. (Important information about using coiled tubing is given in Section 4.7). A possible alternative to coiled tubing, which potentially offers many advantages, is to use an umbilical. This is considered in Section 4.8.

Apart from conveying the cutting fluid and transporting the cutting tool into the well, the coiled tubing has other important functions which must also be considered.

Communication with downhole tools such as positioning sensors, manipulator controls and formation logging equipment is essential. This can be achieved electrically, by installing electrical cable or wireline inside the conveying pipe, or by using a downhole valve arrangement which generates mud pulses in order that information is conveyed in binary code (Ref 74). It is important to note that such a valve is likely to have reliability problems if continuously operated on abrasive mud. Power must also be provided to drive the manipulator. This can be achieved by using a small downhole electric motor or by using a small mud motor. However, mud motors are expensive and may have short lives if operated on abrasive mud. An abrasive-free mud supply might have to be provided.

If electric power is required, the conveying pipe must be able to carry electrical cable. Its size would depend on the power required and the number of separate communication links that would be needed. If an abrasive-free hydraulic supply is needed, a means of providing a separate flow path would be required. In addition, the conveying pipe must be strong enough to carry all the downhole tools and must be able to operate in the corrosive and arduous environment found inside a well.

4.7 Coiled Tubing

Coiled tubing is becoming more widely used and accepted by the oil industry and some of its advantages have already been discussed in Section 3.2.2 and 4.2.4. Coiled tubing is available in sizes from 1.0" OD up to 3.5" OD at up to approximately 15,000 ft (4500 m) long. When using coiled tubing, the available lifting capacity must always be considered. Coiled tubing strings can be connected together if lighter reels have to be used. A typical coiled tubing rig is shown in Figure 4.7.1.

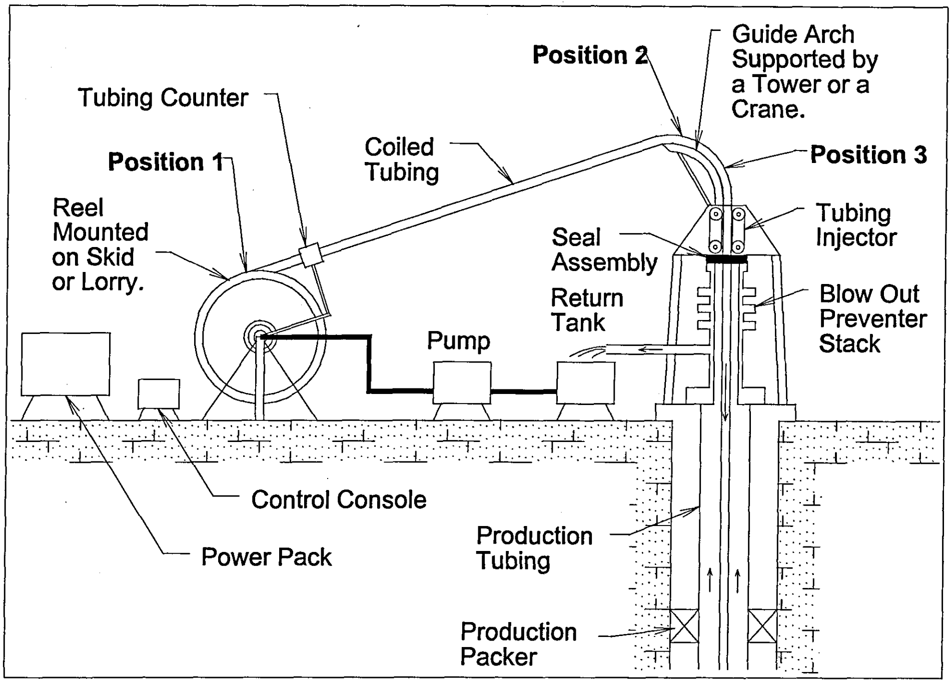


**FIGURE 4.7.1 A COILED TUBING RIG
PLASTIC DEFORMATION POINTS ARE MARKED AT
POSITIONS 1, 2 AND 3.**

4.7.1 Coiled Tubing Life

The coiled tubing has to be plastically deformed to be coiled onto the reel. To use it, it must be plastically deformed a further three times (positions 1, 2 and 3 on Figure 4.7.1): once while being straightened off the reel (1), once while being bent over the guide arch or gooseneck (2), and once while being straightened after the guide arch (3) for injecting into the well. Cycling the tubing into and out of the well therefore imposes considerable bending stresses on the tubing, which results in low cycle fatigue.

The fatigue life is reduced further when operations are conducted with pressure in the tubing (Ref 75). The internal pressure combined with the plastic bending causes the tubing diameter to increase, or balloon, until it has grown so large that the grippers on the injector can no longer clamp the tube.

Table 4.7.1 gives the predicted fatigue life for 2" coiled tubing, at 3 different internal pressures. The values are based on two different wall thicknesses and two grades of steel.

TABLE 4.7.1 TO SHOW THE PREDICTED FATIGUE LIFE FOR 2" COILED TUBING AT DIFFERENT INTERNAL PRESSURES.

Material Grade	Wall Thickness inches	Fatigue Life for Internal Pressure = 0 psi	Fatigue Life for Internal Pressure = 5000 psi	Fatigue Life for Internal Pressure = 10,000 psi
QT 700	0.175	42	26	10
	0.188	42	28	12
QT 800	0.175	52	33	13
	0.188	52	35	16

Note:

Values are predicted for a gooseneck radius of 72" and a reel diameter of 72".

Predictions obtained from Precision Tube Technology.

The fatigue life is given in cycles, where one cycle corresponds to the tubing being unwound into the well and reeled back up.

The very low fatigue lives for coiled tubing with internal pressure indicate that for abrasive water jetting applications, when high internal pressures are needed, the coiled tubing should be held stationary. If cycling with the internal pressure is unavoidable, then the coiled tubing will have to be considered a consumable, just like the nozzles and abrasive, which would be very expensive.

Predicting and monitoring the life of the coiled tubing is therefore very important. Failure could have major safety and cost implications. Computer fatigue models are now available and a service record is kept for each reel of tubing.

Coiled tubing integrity monitors are gradually being developed. These provide the necessary information for the computer predictions, including the tubing's physical dimensions, weight and tolerances. Pia explains that these monitors are based on either the principles of electromagnetics, ultrasonics, or radiant energy (Ref 76).

Once a section of coiled tubing has reached its predicted fatigue life, the section is cut out and the remaining tubing is carefully welded back together.

Oil service companies continually look to use coiled tubing for more demanding duties. These include using it to reach greater distances from the surface, to pull higher and higher loads, to use higher internal pressures and to have longer fatigue lives. These demands have lead to the use of better grades of steel and the development of composite coiled tubing.

4.7.2 Coiled Tubing and Wireline Cable

Coiled tubing can convey wireline cable. The wireline can be installed in the wall of the tubing (Ref 77), although this has the disadvantage of being susceptible to mechanical and chemical damage, or it can be installed inside the coiled tubing, which provides better protection for the cable. Note that wireline cables can be as large as 0.5" in diameter, so that when installed inside small diameter coiled tubing there is little annular space for the cutting fluid.

There are three methods by which cable can be installed inside coiled tubing (Ref 78):

1) by hanging the coiled tubing in a well and then running in the cable. The coiled tubing and cable are then reeled in together.

2) by laying the coiled tubing horizontal and pumping the cable in.

3) by installing a thin pull cable while the coiled tubing is manufactured. The tubing is then laid horizontal and the pull cable is used to pull the cable through the coiled tubing.

All of these methods are expensive, costing \$15,000 - \$25,000 per installation. Any problem with either the coiled tubing or the cable and the cable has to be removed, the repairs made and then reinstalled.

Newman describes a design for a cable injector which can also be used to remove the cable from the tubing. If such a device proves economical then the same coiled tubing can be used to conduct hydraulic operations (no cable installed) and then operations, such as logging, which require the cable to be installed. Blount (Ref 79) describes the installation and use of wireline cable for logging operations.

4.7.3 Concentric Coiled Tubing

Concentric coiled tubing, usually $1\frac{1}{4}$" OD inside $2\frac{3}{8}$"OD, is beginning to be used with a jet pump in order to remove sand from horizontal wells. (Ref 80). High pressure fluid is pumped through the inner bore and ejected through nozzles in the jet pump, so that sand is entrained into the flow and carried back to the surface up the annulus. No information has been found on whether wireline could also be run with concentric coiled tubing, but this would make the system more complicated and if it was installed inside the inner tube, the annular space for the cutting fluid would be very small.

4.7.4 Umbilicals

Umbilical cable systems are currently being used in subsea applications for carrying any combination of electrical power cables, fibre optic cables and hydraulic hoses. They are available in continuous lengths of up to 5 km, depending on the umbilical diameter. Although used to connect subsea wellhead control sensors and valves to the surface, they have not been widely used inside wells. Interest is, however, beginning to develop in their possible use downhole (Ref 80). Current umbilical hose pressure limits are approximately 690 bar for a $\frac{1}{2}$"ID hose, although higher pressure hoses are available.

For an umbilical to be used downhole, considerable development work would be required to overcome several major problems:

1) Existing hoses have an approximate working temperature limit of 50°C-70°C. If a geothermal gradient of approximately 3.6°C per 100 m in depth is assumed (Ref 39), then such umbilicals would be of use down to 1000 m - 1500 m (assuming a 15°C surface temperature). By using fluoro-polymer linings, operating temperatures of up to 120°C may be possible, although the material is very difficult to extrude into long lengths and consequently sections of hose would need to be joined together inside the umbilical (Ref 81).

2) The umbilical components need to be able to withstand the high pressures found inside the well.

3) Special end fittings and internal connectors would need to be designed, particularly if shorter lengths of high pressure hose have to be connected together.

4) All the materials in the umbilical would have to be resistant to the corrosive nature of the oil well. For example, Duplex steels would have to be used instead of some of the more common stainless steels. Strength members, such as kevlar cables, would be required to carry both the weight of the umbilical and the weight of the tools on the end. Armoured sheathing would be needed to protect the exterior of the umbilical from wear.

5) The high capital cost of the umbilical means that its working life must be long enough for it to be economic. The umbilical will be subjected to the same fatigue cycles that are experienced by coiled tubing. Consequently, the more complex umbilical must have a better fatigue life than coiled tubing.

6) The maximum continuous length of umbilical that can be made is roughly inversely proportional to the diameter of the umbilical. This is an important factor when the umbilical is being designed. (JDR cables quoted that for umbilicals up to 50 mm in diameter, 5 kms could be made in one continuous length, whereas only 2 kms could be made for an umbilical of 75 mm in diameter. Ref 82)

The flexibility that an umbilical system potentially offers is very attractive. Several hoses could be used to supply the high pressure cutting fluid and if necessary abrasive-free fluid hoses, electrical cables and even fibre optic cables could also be incorporated. Potentially higher pressure hoses (>1400 bar) could be used, although the consequent reduction in the volume of cutting fluid and the difficulties of getting such a high pressure system accepted by the oil industry could be a problem.

Solutions already exist for most of the points described above, but the high cost of developing a complete umbilical would only be feasible if it could be proved that sufficient advantage could be obtained by using it.

For the purposes of this project, coiled tubing is the only feasible method of conveying the cutting tool and cutting fluid.

4.8 The Basic Requirements of the Cutting Fluid

The cutting fluid must not only carry the abrasive but must also provide all of the properties found in typical drilling fluids. Specifically the cutting fluid must:

1) be able to suspend the abrasive particles, even when there is no flow. Typically the abrasive concentration is 10% by weight or 4% by volume.

2) have a low pressure loss characteristic when flowing.

3) be able to carry and suspend the cuttings and the abrasive back to the surface at low annular velocities. McReynolds recommends a maximum cuttings concentration of 12% when using coiled tubing (Ref 83).

4) if necessary, be weighted to give the balanced or underbalanced conditions required (see Appendix C).

5) contain suitable additives to prevent formation damage and fluid loss and be environmentally acceptable.

Drilling fluids are generally classified as water-based (the most common), oil based, synthetic based or air, mist, foam or gas based (Ref 84). Additives are then mixed to give the desired properties. These include flocculants and viscosifiers, such as bentonite and polymers, which increase the fluid viscosity for better hole cleaning and solids suspension, and weighting materials, such as barite, which have a high specific gravity and are used to control formation pressures. Friction reducers can be added to minimise pressure losses; 2 to 3 times the volume can be pumped when they are used.

Detailed descriptions of the most important considerations for 'building' a drilling mud are given in the IDF Technical Manual (Ref 85). In the end, a compromise has to be made to meet all of the fluid requirements for a particular operation.

How a DIAJET would perform using such a cutting fluid is unknown and would have to be tested. If it does not operate satisfactory then a separate, high pressure water supply might be necessary for the DIAJET, the high pressure abrasive slurry from the DIAJET then being mixed with the high pressure cutting fluid. Alternatively, considering that some oil well pumps are designed to pump abrasive fluids, the DIAJET principal could be adapted to meter abrasive using low pressure water which could then be mixed with the cutting fluid upstream of the high pressure pumps.

4.8.1 Hole Cleaning

Section 3.6 explained that hole cleaning was of particular concern. All of the abrasive and cuttings must be removed from the well, otherwise they could accumulate around the cutting tool and possibly trap it in the well. This would be unacceptable.

In order to select the correct fluid for hole cleaning, the particle slip velocity needs to be calculated. This depends on the size of the cuttings that are produced during abrasive water jet cutting.

Particle analysis has shown that when cutting reasonably homogenous materials the size of the cuttings is no larger than the size of the abrasive used. The only noticeable difference occurs when a concrete type material is being cut, in which fragments of very hard materials, such as flint, are held in the softer cement matrix. The abrasive jet then erodes around the hard particles rather than cutting them and consequently larger cuttings may be obtained.

If solids settling is to be avoided then the upwards, annular velocity of the fluid must be greater than the particle slip velocity. For good hole cleaning McReynolds (Ref 83) explained that the particle terminal velocity, calculated according to Stoke's theorem, must be less than half the annular velocity.

New drilling fluids are designed to stop the particle settling when the annular velocity is low or even when circulation is stopped. At high velocities, the fluid is free to flow without experiencing the high pressure losses associated with highly viscous fluids. One example is a Mixed Metal Hydroxide fluid (MMH). A fluid which exhibits this property is said to be non-Newtonian.

4.8.2 Recommended Base Cutting Fluid

Drilling fluid suppliers (Ref 86) recommended that the following 'base' fluids should be suitable in order to suspend the abrasives and carry the abrasive and cuttings out of the hole at low annular velocities. Specific additives can then be used as required.

Low specific gravity (SG)fluid:

MMH based mud with water, bentonite and caustic soda (inhibits formation clay and shale swelling).

Barite can be added to weight the fluid up to 1.50 SG.

Higher densities might be possible (up to 2.15 SG), although the density is limited by the problem of pumping solids in coiled tubing, by the increase in pressure loss that would be incurred and by the possible interference of the abrasive by the solid weighting particles. This can only be confirmed by testing.

Higher specific gravity fluid:

Brine-based fluid with sodium or potassium formate as the base depending on the density required. Maximum possible SG is 1.598.

The only method of determining if these fluids are suitable is to test them.

In practise, a suitable fluid would be designed and built with the optimum balance between the pressure loss characteristics and hole cleaning characteristics to give the maximum circulation rate and adequate annular velocity.

4.9 Essential Coiled Tubing Tools

If coiled tubing deployed tools are to be used downhole, several standard components must be included in the bottomhole assembly configuration. The most important of these are:

1) Connector Sub
2) Back Pressure Safety Valve
3) Emergency Disconnect
4) Tubing End Locator
5) Centralizer.

The following sections give a basic description of these tools.

4.9.1 Connector Sub

This enables the tools to be connected to the end of the coiled tubing. The connection must be as strong as the tensile rating of the tubing itself and provide a pressure-tight seal, as most operations require high pressure fluid to pass down through the coiled tubing. (Ref 87)

Coronado (Ref 88) describes a slip type anchor which consists of a split slip and a bowl design which anchors the tool string securely to the end of the coiled tubing. O-ring seals are used to provide the pressure seal.

Alternative methods include using setting screws and flaring the end of the coiled tubing (Ref 89).

The restriction of a maximum manipulator diameter of 2" also applies to the connector sub. Consequently the maximum OD of the coiled tubing that can be used is 1.5".

4.9.2 Back Pressure Safety Valve

Before entering the well, the coiled tubing passes through a blow-out preventer which stops it being forced out of the well if there is a pressure surge, or kick, downhole. However, a leakage path could develop if the coiled tubing failed at the surface. This is most likely to occur in the vicinity of the gooseneck. To prevent flow up through the coiled tubing in the event of such a failure, a Back Pressure Safety Valve must be incorporated in the tool string.

The valve is positioned above the Emergency Disconnect so that the tubing is still protected if the disconnect has been activated and to enable circulation through the coiled tubing to continue if required.

4.9.3 Emergency Disconnect Tool

An Emergency Disconnect (ED) must always be incorporated in the tool. This is to ensure that the coiled tubing can be removed from the well if the bottomhole assembly becomes stuck and efforts to free it have failed.

Three types of ED are available; pull disconnect, pressure disconnect and hydraulic or electrical disconnect:

Pull Disconnect

This is designed so that at a predetermined load, applied through the coiled tubing, the connection is sheared apart (Ref 61 and 88). This tool has limited use because it cannot usually be run with other tools that require pull to operate them.

Hydraulic Disconnect

To activate a hydraulic disconnect, a ball is pumped down the tubing until it becomes stuck on an internal sleeve inside the disconnect tool. Hydraulic pressure is then applied. The pressure force acting on the ball loads a set of shear pins, which shear at a certain pressure. Then the internal sleeve moves downwards, releasing the coiled tubing from the remainder of the tool string.

Hydraulic or Electrical Disconnect

When using coiled tubing it is important to have as much pull to the BHA as possible. Using shear devices in the emergency disconnect tool limit the available pulling force. An alternative emergency disconnect tool is discussed in Reference 90. This is controlled by a hydraulic or electrical signal from the surface. On activation, a set of releasing dogs in the upper half of the tool will disengage from the lower sub.

Once released, circulation through the coiled tubing is resumed. This is detected by the operator and well control operations can then be performed while pulling out of the hole. Fishing operations can then be undertaken to try and relieve the stuck tools (Ref 91)

In all of the designs, when the disconnect has been activated and the coiled tubing removed, a profile is left at the top of the stuck tool, called a fish, that can be latched onto easily by fishing tools (Ref 89). Fishing tools available include Hydraulic Jars and Accelerators, which provide impact forces to dislodge the fish, and Overshots and Spears, which catch the end of the fish enabling it to be pulled to the surface.

4.9.4 Tubing End Locator

The depth indicator on the coiled tubing unit is generally not accurate enough to measure the position of the bottomhole assembly. A positive means of depth control is achieved by running a tubing end locator which locates the landing nipple in the end of the production tubing. The measured depth of the nipple is compared with a wireline log of the well and the depth indicator is corrected.

4.9.5 Centralizer

The Centralizer ensures tools are centred in the hole and absorbs side loading, which reduces wear and possible damage to the other tools.

4.10 Summary

This chapter has reviewed existing methods of cutting a window and has shown that a Through-tubing Abrasive Water Jet Window Cutting Tool has considerable potential. Important aspects of the final system have been considered and these will be addressed in the next sections, particularly during the manipulator design stage.

A feasibility study for a Through-tubing Abrasive Water Jet Window Cutting System is presented in the next chapter.

5.0 FEASIBILITY STUDY FOR AN ABRASIVE WATER JET WINDOW CUTTING SYSTEM

In the last chapter a through-tubing window cutting system was identified as a potential application for a downhole abrasive water jet cutting tool. Before a method of manipulating the cutting nozzle(s) can be considered, a feasibility study must be conducted to show that, in theory at least, an abrasive water jet tool can satisfy the cutting requirements. The most important are given in Sections 3.6 and 4.5.2.

The feasibility study must address many different parameters, ranging from the hydraulic factors that effect the final cutting performance to the more practical considerations associated with deploying the tool inside an oil well. These parameters, which are listed below, are discussed in detail in the following sections of this chapter.

Parameters that affect the feasibility of an abrasive water jet window cutting tool include:

- the window cutting strategy.
- the available power for downhole cutting operations.
- optimisation of the nozzle configuration.
- deploying a tool inside an oil well.

If an abrasive water jet window cutting system can be shown to be feasible then a specification can be written for the manipulator and possible manipulator designs can be investigated.

5.1 Cutting Trials on High Strength Steels

Most of the cutting trials conducted at DIAJET Limited have involved cutting stainless steel (304L or 316), Brass or Aluminium. Few cuts have been made on the higher strength steels which are typically used for casing and tubing in the oil industry. The cutting model predicts that these high strength metals should be more difficult to cut. If this is the case, then further correction factors would have to be applied to the downhole cutting predictions.

A selection of standard steel plates used in the oil industry were obtained. A 0.5 mm diameter nozzle at 690 bar using 15% olivine abrasive was used for the trials. By making linear cuts at different traverse speeds the maximum speed to give a parting cut was determined. Table 5.1.1 lists the steels used, their material properties, the predicted parting speed and the actual parting speed measured. For comparison, the predicted parting speed for the same thickness of stainless steel 304L is also given.

The Table shows that the high strength steels tested are as easy or easier to cut than stainless steel 304L.

All further cutting predictions can therefore be based on 304L results; we do not have to worry about the actual type of metal being cut. In practice, a trial cut is recommended on the particular steel being cut, just to confirm the steel follows this general trend.

TABLE 5.1 PARTING CUT SPEEDS FOR A SELECTION OF HIGH STRENGTH OIL FIELD STEELS

Steel	T mm	σ_y MN/m^2	σ_{UTS} MN/m^2	E GN/m^2	Predicted PS mm/min	Actual PS mm/min	Predict PS 304L mm/min
RQT 501	50	560-575	655-675	206	3	12.5	12
50D	50	421	497	206	7	12.5	12
55F	50	503	588	206	6	12.5	12
EN80	9	552	689	206	130	200	200
17-4PH	55	750	1000	197	No cut. (Max depth 33 mm)	15	13

Where:

T	Steel thickness	σ_y	Yield stress
σ_{UTS}	Ultimate tensile strength	E	Young's modulus
PS	Parting speed		

The material properties for 304L are:

$$\sigma_y = 207\ MN/m^2 \qquad \sigma_{UTS} = 552\ MN/m^2 \qquad E = 193\ GN/m^2$$

5.2 Window Cutting Strategy

Different cutting strategies have been considered for cutting a window. While examining these strategies it is important to consider how the cuttings are to be removed from the well, and how the possibility of the cutting tool becoming stuck downhole can be minimised. The strategies generally fall into two categories:

1) the jet traverses over the whole surface area of the window. Small cuttings are then produced (approximately the size of the abrasive particles), which should be easy to remove, although the process is slow. This will be called the 'Zig-Zag' method.

2) the window area is cut into chunks of metal. Unfortunately, these chunks would be the thickness of the casing wall (in the order of 10 mm) and would be too large to be removed from the well by the cutting fluid. In vertical wells it might be acceptable for them to fall to the bottom, but in horizontal wells all of the chunks would have to be removed to prevent the tool becoming trapped.

   A mechanical method of removing them would therefore be required, although collecting the chunks during cutting might be very complicated.

A series of tests were conducted to investigate the feasibility of these cutting strategies.

5.2.1 Test Sample

A crude test sample to simulate cemented casing was made from a 10 mm thick stainless steel plate and a 20 mm thick mild steel plate, with 40 mm of adhesive tile grout holding them together. Cuts were made in the stainless steel using a 1.0 mm diameter nozzle at 200 bar, approximately 15% concentration and a stand-off of 5 - 7 mm. The nozzle movement was controlled by a computer controlled X-Y table.

5.2.2 Zig-Zag Tests

The nozzle was traversed across the workpiece in the zig-zag pattern shown in Figure 5.2.1 at 60 mm/min (note that the parting speed on stainless steel for this nozzle configuration is 75 mm/min). The overlap distance between successive passes was varied from 0.5 mm to 1.5 mm to determine the maximum overlap which would still ensure all of the metal was removed.

Apart from the 1.5 mm separation, all of the steel was cut and all of the grout behind the steel was removed. No careful control of the traverse speed was required.

5.2.3 Conclusion of Zig-Zag Tests

If an overlap distance equal to the nozzle diameter is used, all metal should be removed without a problem.

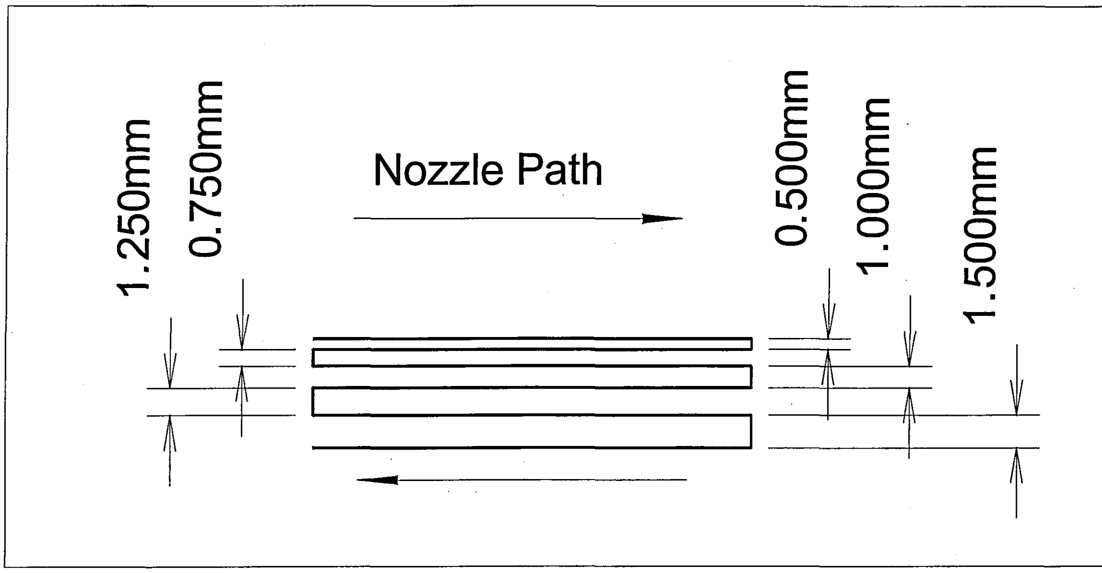


FIGURE 5.2.1 ZIG-ZAG NOZZLE PATH

5.2.4 Cutting The Window into Pieces

The nozzle followed the path shown in Figure 5.2.2, so that two squares of metal should be removed. In order to cut the squares, cut paths must intersect each other. It was at these points of intersection that metal was found to be left uncut.

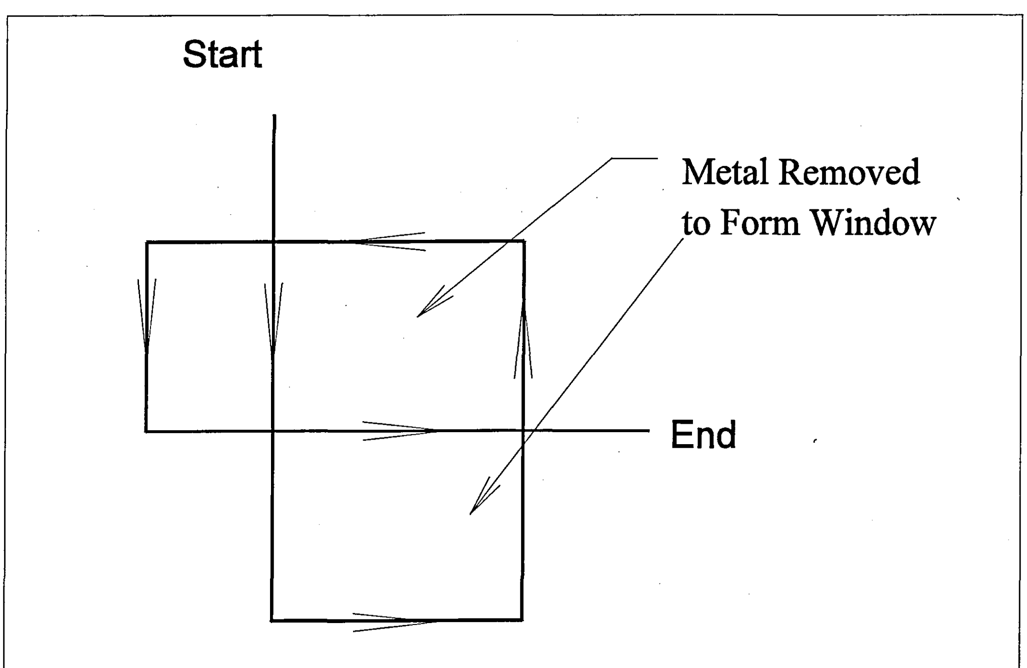


FIGURE 5.2.2 NOZZLE PATH TO CUT A WINDOW INTO CHUNKS

As the jet cuts through the metal it slows and deflects backwards (see Figure 5.2.3 and explanation Section 2.4). Consequently, when the top of the jet reaches a previously cut section, it jumps into the gap and leaves a sliver of uncut metal at the bottom of the cut.

A similar problem was encountered at the corners of the squares, if the jet was moving too quickly. When the nozzle reaches the corner and changes direction, the deflected jet at the bottom of the cut, which is still moving into the corner, does not have time to catch up with the jet at the top of the cut, which is moving away from the corner. Consequently, metal is left uncut in the corners.

Slowing the traverse speed from 50 mm/min to 20 mm/min reduced the amount of metal left uncut, but this was not reliable. In some cases, the squares were removed but sharp protrusions were left where the metal was forced out rather than cut.

With the sophisticated control of the X-Y table, it was possible to slow the traverse speed only in the regions of the previous cuts and the corners. Then all the metal was removed. However, it is likely that such complicated control could not be reliably achieved remotely inside an oil well.

An alternative was to traverse over the cut a second time, either in the same direction as the first pass or in the opposite direction. Cutting in the opposite direction was found to be more reliable, with all of the metal being cut at a traverse speed of 50 mm/min.

For these cuts, all of the grout behind the squares was washed away. The surface of the lower, mild steel plate was only scratched where the nozzle was either stationary or traversing very slowly.

5.2.5 Conclusion of Cutting the Window into Pieces

These tests indicate that cutting a window into chunks cannot be reliably achieved. Considering that it is unlikely that such chunks could be reliably removed from the well, this cutting strategy must be abandoned.

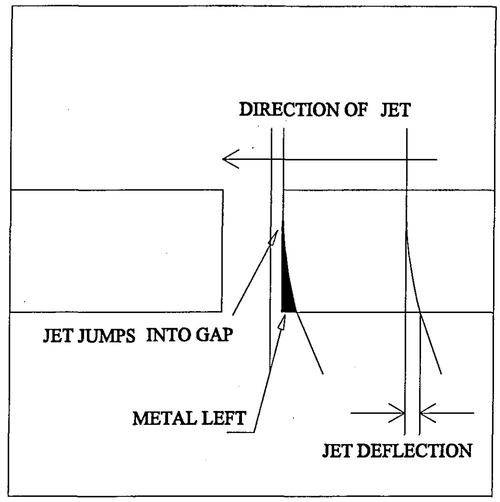


FIGURE 5.2.3 THE CUTTING PROCESS

5.2.6 Overall Conclusions of the Window Cutting Tests

The only reliable method of cutting a window with an abrasive water jet is to traverse the nozzle over the whole surface area of the window. Accurate nozzle control is particularly important as the distance between successive cuts, equal to or less than the nozzle diameter, must be maintained to ensure all metal is cut.

This method will only be suitable if the possible nozzle configuration and the size of the window enable cutting to be completed in a comparable time to conventional techniques.

Section 4.5.2 describes a typical window as 2 - 2.5 m long and 3.75" wide. Assuming a casing thickness of 10 mm, a nozzle configuration must be selected to cut 20 mm thick stainless steel, to account for the reduction in cutting performance at back pressure. If we assume a 2.0 mm diameter nozzle can be supplied with 10% abrasive cutting fluid at 300 bar, then a traverse speed of 80 mm/min will be possible. The time to cut the window, assuming a regular Zig-Zag pattern, is then:

$$= \frac{\textit{Length of Window}}{\textit{Traverse Speed}} \cdot \frac{\textit{Width of Window}}{\textit{Zig-Zag Separation}}$$
$$= \frac{2000}{80} \cdot \frac{3.75 \,.\, 25.4}{2}$$
$$= 1190 \ \textit{minutes}$$
$$= 19 \ \textit{hours.}$$

This is considerably longer than the 10 hours typically quoted for conventional methods. However, if two nozzles could be used then the cutting time would be comparable.

In the end, window cutting may only be viable for particular well configurations, where the required volumes and pressures can be supplied to the nozzles.

5.3 Available Power for Downhole Cutting Operations

Based on the available information, estimates need to be made of the hydraulic power that can be conveyed to the nozzle through the coiled tubing.

Several factors affect the final hydraulic power that is available for cutting and these are considered below.

5.3.1 Wear of Coiled Tubing

In order to minimise the wear of the coiled tubing due to erosion by the abrasive fluid, a velocity limit is imposed on the cutting fluid. Extensive work has been conducted on the wear effects of slurry transport (Ref 92) and velocities of less than 3 m/s for continuous duty have been found to give acceptable wear rates.

Wear due to erosion is unlikely to be a problem because of the low fatigue life of the coiled tubing. Erosion only needs to be considered in the region of the manipulator or other components, which experience little or no bending fatigue. Protective coatings may then be needed if local velocities are higher than 3 m/s.

5.3.2 Pressure Losses in Coiled Tubing

As the cutting fluid flows down the coiled tubing, friction between the fluid and the tubing walls results in a pressure loss. This loss should be minimised to ensure the maximum pressure drop is available across the nozzle in order to accelerate the abrasive particles to the maximum speed possible. The magnitude of the pressure losses will depend on whether the cutting fluid is Newtonian or non-Newtonian. To give an indication of the pressure losses, the cutting fluid carrying the abrasive will be assumed to be plain water (Newtonian fluid).

The general equation for the pressure loss associated with a fluid flowing in a pipe is:

$$\Delta P = f\left(\frac{L}{d}\right)\rho\left(\frac{u^2}{2}\right)$$

where:

ΔP= pressure loss in pipe (Pa)
f= friction factor
L= length of the pipe (m)
d= internal diameter of pipe (m)
u= fluid velocity in pipe (m/s)
ρ= fluid density (kg/m^3)

The friction factor depends on whether the flow is turbulent or laminar, on the relative roughness of the pipe and on its cross-sectional shape.

The flow condition inside the pipe can be predicted by calculating the Reynolds Number, Re, where:

$$Re = \frac{\rho u d}{\mu}$$

μ= fluid dynamic viscosity

Flow conditions:

$Re < 2000$ - Laminar.
$2000 < Re < 5000$ - Transition
$Re > 5000$ - Turbulent.

Laminar flows usually develop in the coiled tubing - casing annulus and turbulent flows in the coiled tubing.

For flows in a concentric annulus, the hydraulic diameter is used to calculate the required coefficients (Ref 93). The hydraulic diameter = (d_2 - d_1), where d_1 and d_2 are the inner and outer annulus diameters respectively.

Further correction factors can be applied to account for eccentric annuli, but these will not be considered as the highest pressure loss corresponds to a concentric annulus.

Laminar Conditions

For laminar conditions the friction factor is given by:

$$f = \frac{Cf}{Re}$$

where $Cf = 64$ for circular cross-sections.

For annular cross-sections, Cf depends on the ratio $\frac{d_1}{d_2}$.

For $\frac{d_1}{d_2} > 0.2$, Cf varies between 92 and 96.

For simplicity, Cf is assumed to be 96 for all annular flows.

Turbulent Conditions

For turbulent conditions the friction factor can be calculated using the Moody chart, which is a plot of the Colebrook-White equation, or from the empirical equation below, which is more convenient for computer calculations (Ref 93):

$$f = \frac{0.25}{\left(\log_{10}\left[\frac{e}{3.7d} + \frac{5.74}{Re^{0.9}}\right]\right)^2}$$

where:

$e =$ wall roughness (m), which depends on the pipe material, the method of manufacture and the deterioration of the surface over time.

The annular friction factor, $f_{\text{annular}} = 1.05\, f_{\text{circular}}$.

Some typical roughness values for steel pipes are:

New smooth pipes	$= 2.5*10^{-5}$ m
Light rust	$= 2.5*10^{-4}$ m
Heavy rust	$= 1.0*10^{-3}$ m

5.3.4 Examples of Pressure Losses in Coiled Tubing

Using this analysis and assuming flow velocities of 2 m/s and 3 m/s, pressure losses per 1000 m length of coiled tubing were calculated (Table 5.3.1). Note that the first pressure loss value corresponds to a velocity of 2 m/s.

TABLE 5.3.1 PRESSURE LOSSES IN SELECTED COILED TUBING

Pressure Limit bar	OD inches	ID mm	Area X-sec m^2	Vol l/min (2 m/s)	Vol l/min (3 m/s)	ΔP per 1000 m bar
786	1.00	19.9	$3.1*10^{-4}$	37	56	43-97
834	1.50	29.2	$6.7*10^{-4}$	80	121	26-57
675	2.00	41.2	$1.3*10^{-3}$	160	240	16-36
597	2.38	50.7	$2.0*10^{-3}$	242	363	12-28

Coiled tubing (CT) information obtained from Precision Tubing Technology catalogue.

Note: Hydro Test pressure capacities are given for the coiled tubing. This test pressure value is 80% of the Burst Yield Pressure Rating, which is the internal pressure to cause yielding using the minimum yield strength and the minimum wall thickness.

Pipe roughness = $2.5*10^{-4}$ m, corresponding to a condition of light rust.
ρ=1000 kg/m^3
μ=1*10^{-3} kg/ms

Note: flowing at 2 m/s, the cutting fluid would take 8.3 minutes to reach a nozzle 1000 m away.

Table 5.3.2 gives the pressure loss for a flow velocity of 2 m/s and 3 m/s for a 0.5" wireline installed inside the selected coiled tubing.

TABLE 5.3.2 PRESSURE LOSSES IN SELECTED COILED TUBING WITH 0.5" WIRELINE INSTALLED INSIDE

Pressure Limit bar	OD inches	ID mm	Annulus area m^2	Vol l/min (2 m/s)	Vol l/min (3 m/s)	ΔP per 1000 m bar
786	1.00	19.9	$1.8*10^{-4}$	21.6	32.4	181-401
834	1.50	29.2	$5.4*10^{-4}$	64.8	97.2	57-128
675	2.00	41.2	$1.2*10^{-3}$	144	216	28-62

The tables show the importance of the flow velocity on the pressure loss. Although 3 m/s was defined as the maximum allowable velocity to minimise wear, a velocity of 2 m/s is more realistic in order to minimise the pressure loss. Lower velocities could be used, but then the volume of fluid would be insufficient to run a large nozzle and the pumping time for the fluid to reach the nozzle might be unacceptable.

If wireline is then installed in the coiled tubing, the reduction in the volume of the cutting fluid and the increase in the pressure loss is significant. Consequently, although the pressure rating is lower, coiled tubing of 1.5"OD is preferable. This rules out the use of wireline with concentric coiled tubing.

Similar pressure loss calculations must be conducted for the coiled tubing - casing annulus. Once all of the pressure losses have been calculated, the remaining pressure at the nozzle can be found.

Note that because of the worry of coiled tubing fatigue failure, large safety factors must be applied to the coiled tubing pressure ratings.

5.3.5 The Allowable Working Pressure

The maximum pressure usually used on an oil well, except in special circumstances, is 5000 psi (345 bar), even though equipment is rated to over 10,000 psi (700 bar). No maximum working pressure limits are defined in the Health and Safety Executive Standards for offshore equipment. High pressure equipment therefore has to comply to existing standards for high pressure technology, such as the High Pressure Safety Code published by the High Pressure Technology Association.

The maximum working pressure of 5000 psi appears to be imposed by the oil industry, who generally seem apprehensive at using higher pressures. If a high pressure abrasive jet cutting system is therefore going to be accepted, this problem will have to be addressed by demonstrating to the clients the safety of the system and by carefully instructing the operators how to use it correctly.

Section 5.2.1 showed that steel could be cut with a nozzle pressure of 200 bar. Assuming a maximum surface pressure of 350 bar (5000 psi), then up to 150 bar can be lost due to friction. Assuming that the coiled tubing pressure loss is approximately 60 bar (see Table 5.3.1) and that the pressure loss for the coiled tubing - casing annulus is negligible, then an abrasive water jet cutting tool can be used up to 2500 m from the surface. This is greater than the average well depth discussed in Section 4.5.1. Ideally the highest possible surface pressure should be used, to maximise the potential operating distance from the surface and cutting speed.

Polymer additives can reduce the friction loss by as much as a factor of 3, which would give a maximum operating distance of 7500 m from the surface.

5.4 Optimisation of the Nozzle Configuration

5.4.1 Spreadsheet Calculation Program

A spreadsheet program was written which calculates the optimum nozzle configuration to cut a window in the minimum time. This is based on the basic pressure loss equations and the cutting data obtained for stainless steel 304L.

Several assumptions had to be made in the program. These are:

1) The pressure losses are calculated for a Newtonian fluid rather than the non-Newtonian fluid that would actually be used. This is because little detailed information is available on the drilling fluid that would be used. Calculations for Newtonian fluids should also give the minimum possible window cutting performance.

   Non-Newtonian calculations based on the equations defined in the IDF Technical Manual (Ref 85) can easily be incorporated into the spreadsheet. Their settling velocity calculations have compared well with experiments conducted on CMC fluids at BHR Group.

2) Not all of the coiled tubing will be unwound off the reel. However, the calculations do not include the pressure loss associated with the tubing left on the reel. This pressure loss will be higher than if the coiled tubing was straight.

3) For simplicity, the annular pressure loss between the coiled tubing and the production tubing is calculated for the full length of the coiled tubing in the well. In practice it is likely that a considerable distance could be between the end of the production tubing and the point where the window is cut. The annular pressure loss between the coiled tubing and the casing is usually negligible compared to the pressure loss between the coiled tubing and the production tubing, so this assumption will over estimate the pressure loss.

There are a considerable number of variables involved in calculating the nozzle configuration. In order to present meaningful trends, several have to be assumed. The calculation must then be repeated if one of these variable has to be changed. The assumed variables are listed below.

Assumed Variables

1) Production tubing size of $2\frac{7}{8}$". This will limit the maximum OD of the coiled tubing and manipulator.

2) Coiled tubing size. Typically $1\frac{1}{2}$" and $1\frac{3}{4}$".

   For efficient cutting, high pressures will be required. Consequently, coiled tubing with maximum wall thicknesses will be used.

3) If a wireline cable is installed inside the coiled tubing, the wireline diameter is needed. In general this is (Ref 94):

   $\frac{7}{32}$" or $\frac{5}{16}$" monocable,

   $\frac{5}{16}$" coaxial,

   $\frac{7}{16}$" 7 conductor,

   although other sizes are available.

4) Casing size and wall thickness to be cut.

   Typical casing will be 5" and 7" in diameter with wall thicknesses in the order of 10 mm.

   To account for the reduction in cutting performance due to the jet being submerged at depth, the required thickness to be cut is doubled.

5) The length of coiled tubing required to reach the window.

6) The coiled tubing pressure limit or, if smaller, the maximum surface pressure that can be used (see Section 5.3.5). A large safety factor would normally be applied to account for the age and usage of the coiled tubing (20% has been used for this example).

7) The cutting fluid is assumed to be water with a density of 1000 kg/m^3 and a dynamic viscosity of $1*10^{-3}$ kg/ms. As an estimate, increasing the density will give an indication of how the pressure loss might change if a more viscous fluid was used.

8) The window size is initially assumed to be 2.5 m long and 3.75" wide. The exact size will depend on how easily the window can be relocated and accessed by the through-tubing drilling assembly.

9) The number of nozzles used.

10) The formation pressure gradient is needed. If unknown, the well can be assumed to be in a state of balance or the average formation pressure gradient can be used.

5.4.2 The Basis of the Calculation

Fluid Flow Calculations

For a particular number of nozzles, N, and a specific nozzle diameter, d_N, the computer estimates the volume flow rate of cutting fluid required, V. The volume flow rate per nozzle, Q, is then:

$$Q = \frac{V}{N}$$

Using Q, the pressure drop across each nozzle, ΔP_N, is calculated (Appendix A):

$$\Delta P_N = \left(\frac{\rho}{2}\right) \cdot \left(\frac{4 \,.\, Q}{\pi \,.C_d \,.\, d_N^2}\right)^2$$

where:

ρ is the cutting fluid density,
C_d is the coefficient of discharge for the nozzle.

Using V, the pressure losses in the coiled tubing, $\Delta P_{CT,}$ and coiled tubing - casing annulus, ΔP_{Ann} are calculated:

$$\Delta P_{CT} \,, \Delta P_{Ann} = f \left(\frac{L}{d_{eff}}\right) \rho \left(\frac{u^2}{2}\right)$$

where

f depends on whether the flow is laminar or turbulent (see Section 4.3.2),
d_{eff} is the effective coiled tubing or annulus diameter,
u is the fluid velocity in the coiled tubing or coiled tubing-production tubing annulus,
L is the length of coiled tubing.

The pressure difference between the formation pressure and the static head of the cutting fluid is also calculated:

$$\Delta P_{F-SH} = (FPG - \rho \, . \, g) \, . \, h$$

where:

FPG is the Formation Pressure Gradient
(Appendix C gives an average of 10.52 kPa/m),
g is the acceleration due to gravity,
h is the depth of operation.

The total surface pressure, $P_{Surface}$ is then:

$$P_{Surface} = \Delta P_N + \Delta P_{CT} + \Delta P_{Ann} + \Delta P_{F-SH}$$

which is compared to the surface pressure limit, $P_{Surface_{Lim}}$.

The computer then recalculates the total volume of cutting fluid, V, until $P_{Surface_{Lim}} - P_{surface} = 0$.

This then gives the optimum flow rate for the nozzle configuration selected.

<u>Cutting Predictions</u>

20 mm thick stainless steel can be cut at 80 mm/min using a 2.0 mm diameter nozzle at 300 bar and 10% abrasive concentration.

Using this information, and the nozzle pressure drop calculated above, the traverse speed is calculated for the casing thickness that needs to cut (Section 2.4):

$$TS = \left(\frac{\Delta P_N}{300} \right)^{1.5} . \left(\frac{d_N}{2} \right) . \left(\frac{20}{T} \right) . \, 80$$

where:

TS is the traverse speed.
T is the corrected casing thickness to be cut. Note that this is twice the actual casing thickness.

This equation assumes 10% abrasive by weight and that the nozzle stand-off is less than 5 nozzle diameters.

Time Required to Cut the Window

Finally, for a window length L_w and width W_w, the time to cut the window, t, is:

$$t = \left(\frac{L_w}{TS}\right) \cdot \left(\frac{W_w}{d_N}\right) \cdot \left(\frac{1}{N}\right)$$

This assumes a nozzle step on the zig-zag cutting path equal to the nozzle diameter and that, when multiple nozzles are used, the cutting is shared equally between each nozzle.

These calculations are repeated for nozzle sizes between 1 mm and 3 mm in diameter. A graph of nozzle diameter against traverse speed and the time to cut the window is generated, from which the nozzle diameter which can cut the window the quickest can be selected. Associated volume flows and pressures can be read off a separate graph.

This whole procedure must then be repeated for a different number of nozzles, until the fastest configuration is identified.

Note that the more nozzles used, the smaller each nozzle must be. There is then an increased chance of nozzle blockage, as well as a reduced performance at increased stand-off distances. Generally, larger nozzles (>2.0 mm in diameter) should be used to minimise these problems.

5.4.3 Results

Graphs 5.4.1 and 5.4.2 show a typical set of nozzle performance curves for one and two nozzles and the following arrangement:

Depth of Operation 3000 m	Well assumed to be in a state of balance.
Production Tubing OD 2.875"	Production Tubing ID 2.44"
Length of Window 2500 mm	Width of Window 100 mm
Coiled Tubing OD 1.5"	Coiled Tubing ID 1.188"
Surface Pressure Limit 595 bar	0.5" Diameter Wireline Cable Installed.

Accompanying graphs showing the pressure losses and the required volume flows are also given for information.

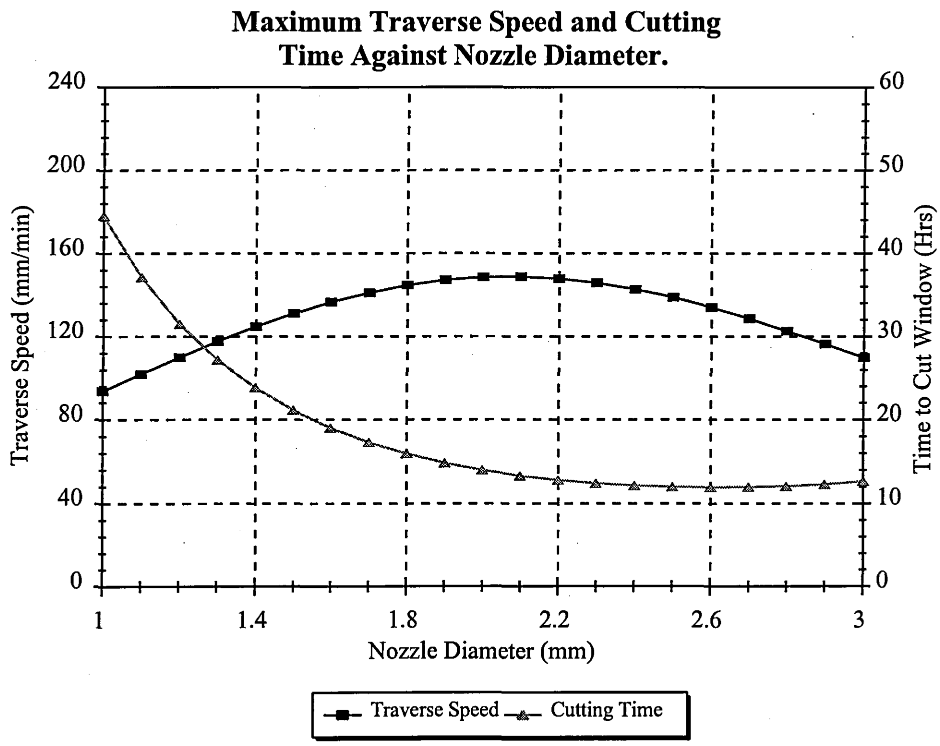
The curves show that to cut a window using **one** nozzle, a 2.6 mm diameter nozzle will be the quickest at cutting. This should cut the window in approximately 12 hours, traversing at a speed of 136 mm/min. Alternatively, two 1.8 mm diameter nozzles traversing at 100 mm/min should cut the window in approximately the same time.

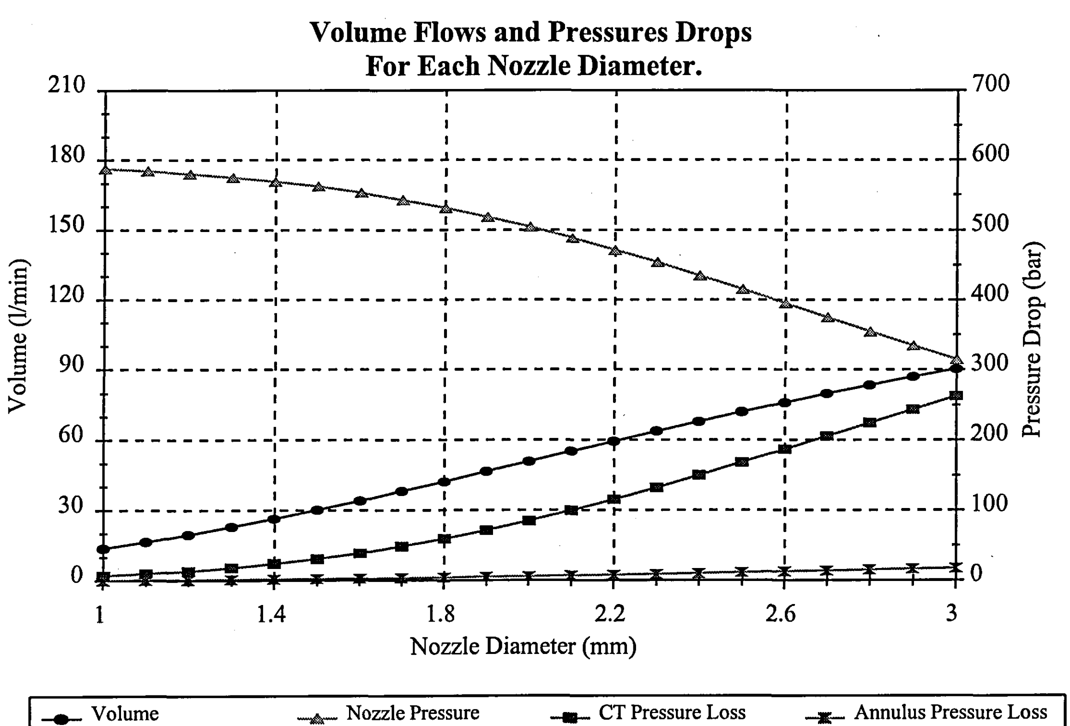
The fact that the same time is required to cut the window with one or two nozzles is not surprising, considering that the same energy is being supplied in each case. However, the 2.6 mm diameter nozzle has a better cutting performance at large stand-off distances and is therefore preferable. For this reason, a 3.0 mm nozzle would be a better selection, although it takes slightly longer to cut the window.

Note that the surface pressure limitation means that above a 2.2 mm single nozzle, the traverse speed starts to decrease.

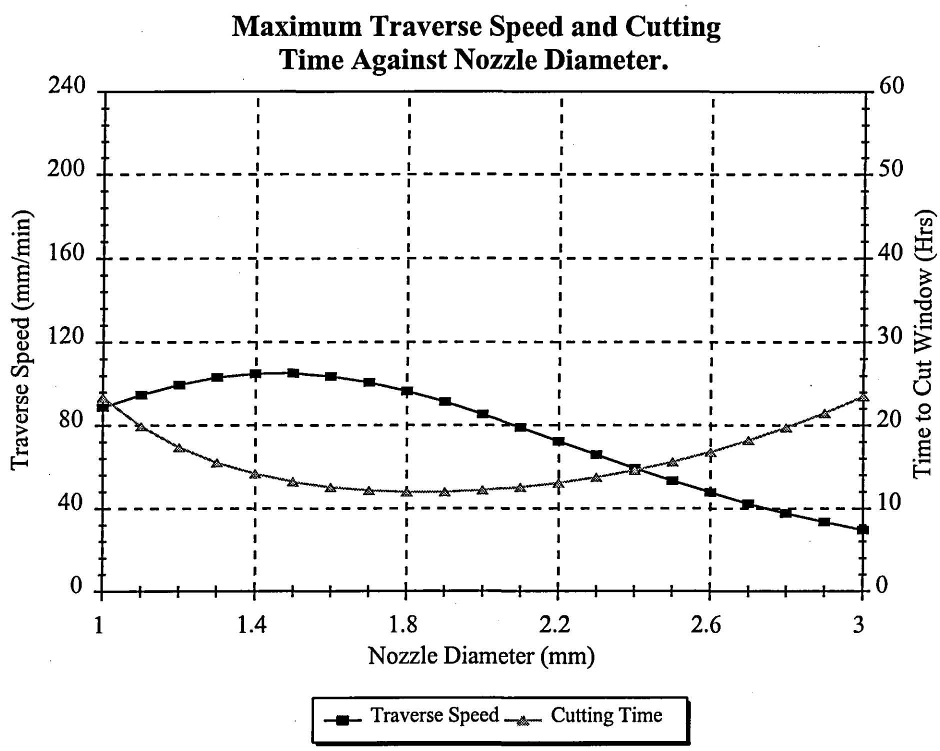
The final selection depends on the manipulator design. For example, by using two nozzles a more compact manipulator may be possible. However, a larger, single nozzle is preferable because of its better cutting performance with stand-off distance. The final positioning of the manipulator in the well will determine how critical this will be.

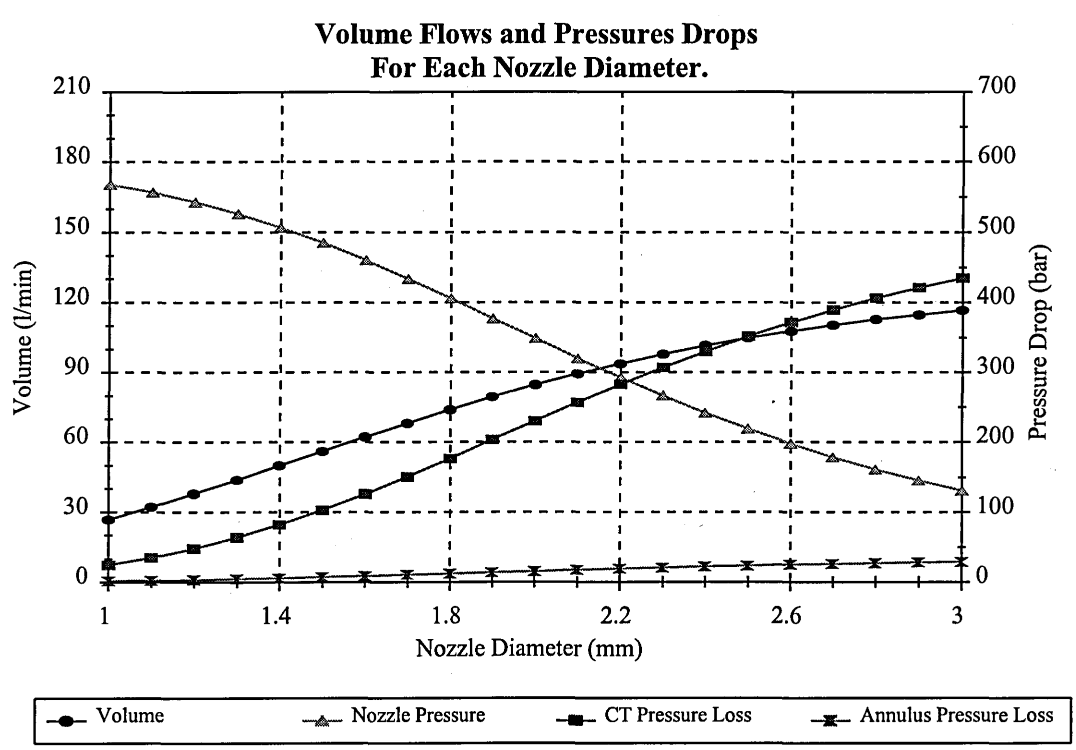
GRAPH 5.4.1 TYPICAL RESULTS FROM THE OPTIMISATION PROGRAM FOR 1 NOZZLE





GRAPH 5.4.2 TYPICAL RESULTS FROM THE OPTIMISATION PROGRAM FOR 2 NOZZLES





5.4.4 Accuracy of the Program Results

Surface Roughness Factor

The selection of the surface roughness factor can have a significant effect on the pressure losses, as shown in Table 5.4.1.

TABLE 5.4.1 TO SHOW THE EFFECT OF VARYING THE SURFACE ROUGHNESS FACTOR BY ± 100% ON THE NOZZLE PERFORMANCE PREDICTIONS

Surface Roughness (m)	$2.5*10^{-4}$	$5*10^{-4}$ (*2)	$1.25*10^{-4}$ (*0.5)
$\Delta P_{L_{CT}}$ (bar)	86	103 (+20%)	74 (-14%)
$\Delta P_{L_{Ann}}$ (bar)	48	58 (+21%)	42 (-14%)
ΔP_N (bar)	444	418 (-9%)	462 (+4%)
Predicted traverse speed (mm/min)	123	112 (-9%)	130 (+6%)

These results were generated by the program for the following basic configuration:

two, 2 mm diameter nozzles,
3000 m operating depth,
1.75" OD, 1.42" ID coiled tubing,
$2\frac{7}{8}$" OD, 2.441" ID production tubing,
ΔP_{F-SH} equal to zero.

Table 5.4.1 shows that a 100% error in the surface roughness will lead to errors in the pressure losses of between 14% and 20%. This equates to a variation in nozzle pressure drop of between 4% and 9% and consequently a variation in the traverse speed required to cut the casing of between 6% and 9%.

The safety factors included in these calculations more than cover this range of error.

Roughness values are generally selected from experience. Sas-Jaworsky (Ref 95) details similar coiled tubing pressure loss calculations. He uses a roughness of $4.5*10^{-5}$ m, corresponding to new steel. However, it is better to use a conservative value rather than be unable to cut the window.

Pipe Diameter

The pipe diameters used in these calculations are also very important. This is because the pressure loss is inversely proportional to the fourth power of the diameter. A 10% error on the coiled tubing ID of 1.42" could lead to a 20% error on the traverse speed required. Table 5.4.2 shows this for the system configuration detailed with Table 5.4.1.

TABLE 5.4.2 THE EFFECT OF A 10% ERROR IN THE COILED TUBING INTERNAL DIAMETER ON THE PERFORMANCE CALCULATIONS

Coiled Tubing ID (inches)	$\Delta P_{L_{CT}}$ (bar)	$\Delta P_{L_{Ann}}$ (bar)	ΔP_N (bar)	Predicted traverse speed (mm/min)
1.42	86	48	444	123
1.278 (-10%)	152 (+77%)	42 (-12%)	384 (-13%)	99 (-20%)

Note:

the roughness used is $2.5*10^{-4}$ m.

the annular pressure loss is different because a new optimum flow rate and nozzle configuration has been calculated.

If the diameter is small, then this error can be magnified even further. Consider the production tubing ID being reduced by 10% from 2.441" to 2.197"; perhaps there has been a section which has collapsed. Table 5.4.3 shows the effect of this change on the performance predictions, again for the basic configuration detailed with Table 5.4.1.

TABLE 5.4.3 THE EFFECT OF A 10% ERROR IN THE PRODUCTION TUBING INTERNAL DIAMETER ON THE PERFORMANCE CALCULATIONS

Production Tubing Diameter (inch)	$\Delta P_{L_{CT}}$ (bar)	$\Delta P_{L_{Ann}}$ (bar)	ΔP_N (bar)	Predicted traverse speed (mm/min)
2.441	86	48	444	123
2.197 (-10%)	66 (-23%)	177 (+370%)	336 (-24%)	81 (-34%)

In this case, the annular pressure loss has increased by 300% and consequently the actual traverse speed required to ensure a complete cut must be reduced by 34%.

The built-in safety factors would only just allow for such a variation. However, by logging the production tubing, accurate diameters can be measured. Generally, any significant changes in diameter will be localised, and therefore not a problem unless they are so significant that tools cannot pass through. In this case the production tubing would have to be removed.

Important Note

The above analysis is for a Newtonian fluid. The addition of friction reducing agents can reduce the pressure loss by up to 70% of the values calculated here (Ref 38). These results therefore represent a worst case and demonstrate the minimum performance of an abrasive water jet cutting tool. Specialist service companies would be required to design the cutting fluid which transports the abrasive. These companies have extensive computer programs which conduct the same pressure loss calculations for non-Newtonian fluids.

5.5 Associated Procedures for Cutting a Window with an Abrasive Water Jet

5.5.1 Positioning the Abrasive Water Jet Window Cutting Tool in the Casing

The conventional methods for positioning a through-tubing milling tool are:

a) cement sidetrack,

b) whipstock in cement,

c) expandable whipstock.

These are described in detail in Section 4.2.4

Of these three methods, an adaptation of the whipstock in cement method should be suitable for positioning the cutting nozzles as close to the casing wall as possible. A cement plug can be placed in the required position in the casing, and a hole directionally drilled through the cement to the casing wall. On reaching the casing wall, the drill bit cuts a cavity alongside and parallel to it. The window cutting tool can then be positioned in the cavity, locking into an anchor placed at the bottom.

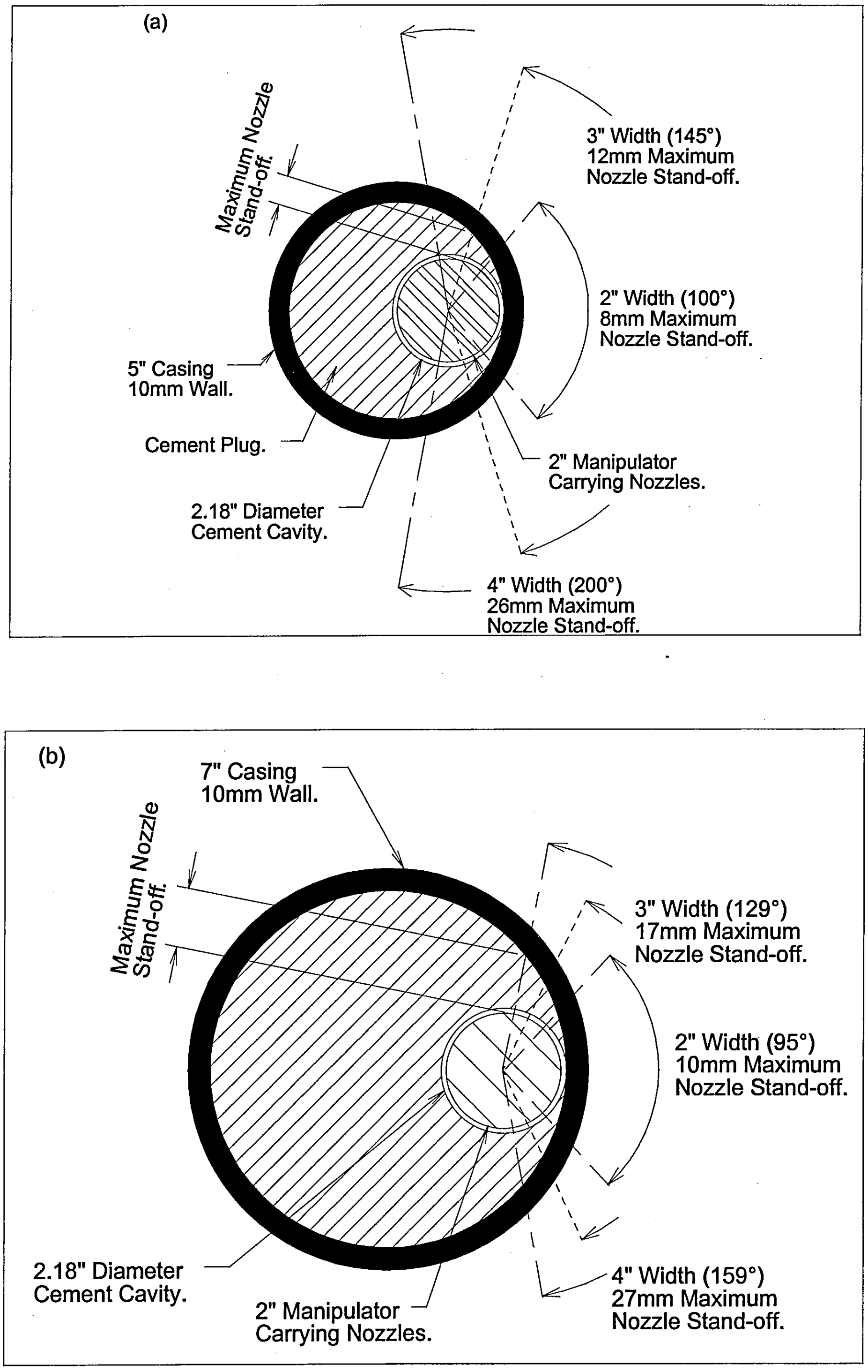
Figures 5.5.1 (a) and (b) show a possible arrangement for a 2" diameter manipulator, which carries the nozzles and traverse mechanism, offset in a 5" and 7" OD casing respectively. The manipulator is positioned in a cement cavity 2.18" in diameter. The diagrams show the maximum nozzle stand-off distance for different widths of window.

For windows less than 3" wide, the maximum stand-off is 12 mm for 5" casing and 17 mm for 7" casing. A 3 mm diameter nozzle should be able to cut, at the predicted performance, at a stand-off of 12 mm (12 mm is less than 5 nozzle diameters). However, for 17 mm the cutting performance will be reduced.

Ideally a method of continually altering the nozzle stand-off distance is required. This is very difficult to achieve reliably. The only alternative is to reduce the traverse speed as the stand-off distance increases.

Note that the nozzle arrangement patented by TIW (Section 4.3.2) allows the stand-off to be reduced, but it cannot be continually varied.

If only a 2" drill bit can enter through the production tubing, then the minimum width of the window must be greater than 2", although the wider the window, the easier the re-entry through it.



FIGURES 5.5.1 (a) and (b)
2" DIAMETER MANIPULATOR POSITIONED IN A 2.18" DIAMETER CEMENT CAVITY, IN 5" AND 7" CASING

The figures show possible widths of windows with the corresponding angle that the nozzle has to be turned through and the maximum nozzle stand-off distance.

5.5.2 Dressing the Completed Window

The nozzle traverse speed is selected to ensure that all metal is removed. Cement and formation behind the casing will also be cut and a wash-out zone generated. This zone will become filled with cutting fluid, abrasive, cuttings, and any formation fluids. If, once cutting is completed, attempts are made to drill through the window, it is likely that the drill bit will take the wrong route in the void and become stuck. Stuck tools are particularly a problem when the tool is being withdrawn through the window.

To avoid this, the void should be cemented. Then the well can be re-entered with a directional drilling assembly and the sidetrack can be drilled through the cemented void and into the formation. See Figure 5.5.2.

If necessary, to help the drill bit locate the window, a whipstock could be placed in the cement cavity before the void is cemented. Provided the whole cavity is not filled, in which case finding the window would be almost impossible, the drill bit will enter the cavity, drill through the new cement and, on reaching the whipstock, will be deflected through the window.

This is similar to the conventional whipstock in cement technique. The only difference is that the metal casing, which is more difficult to cut than cement or formation, has already been removed.

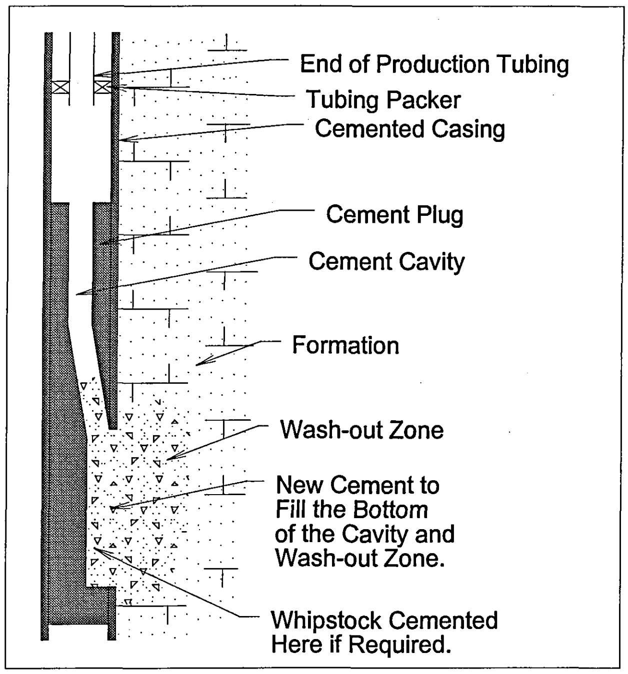


FIGURE 5.5.2 COMPLETED WINDOW AND CEMENTED VOID READY FOR SIDETRACKING

5.5.3 Completing the Sidetrack

Once the window has been cut and the sidetrack drilled, the well must be completed. In conventional drilling, the well would either be cased or left open (see Section 3.3.1). In this situation, because of the restriction imposed by the size of the production tubing, it is likely that the sidetrack would be left open.

If the formation around the sidetrack is unstable the coiled tubing could be left in the well. Pre-perforated coiled tubing may even be available to ensure maximum exposure to the reservoir. Alternatively, an electrochemical perforating tool could be used. In this case, the tool would either be run on wireline or, if the hole angle is too steep to use wireline, smaller diameter coiled tubing. Possible completion options are described in detail by Dickinson (Ref 36)

5.5.4 Swarf and Abrasive

All continuous or long duration cutting operations suffer from the problem of swarf management. This will also be a problem for a window cutting tool.

The cutting fluid will be designed to remove all abrasive and cuttings from the well and ensure that they do not settle if circulation has to be stopped. Section 4.8 covers this in detail.

But what happens to the abrasive after it has hit the target?

Initially, before the jet has pierced the metal, the jet will be reflected back onto the manipulator. The manipulator will be eroded severely if adequate surface protection is not provided.

Once the jet has pierced the metal, it will lose energy cutting into the cement and formation. At some point the jet will have insufficient energy to cut further and will again deflect back. In this case, erosion of the manipulator will be less severe, as the energy of a submerged jet is dissipated rapidly, until the cavity is large enough that the deflected jet does not reach the manipulator.

Any moving parts within the manipulator must therefore be designed in the knowledge that abrasive will be present.

How the abrasive particles are removed from the well, particularly from the region around the manipulator, was not clear on first examination. It is possible that they would be left at the bottom of the cavity, instead of being 'picked up' by the circulating fluid. This would be unacceptable as it increases the chance of the tool becoming stuck downhole. To investigate this, a crude hole cleaning experiment was undertaken.

Hole Cleaning Experiment

A nozzle holder carrying a 2.3 mm diameter nozzle as shown in Figure 5.5.3, was connected to a mains water hose and placed vertically inside a measuring cylinder. The nozzle was then pointing to the wall of the cylinder.

The cylinder was filled with water and some almost neutral buoyancy, 0.5 mm diameter, polystyrene balls were added to the water. (Note that the approximate size of the abrasive particles that would actually be used for cutting with a 3 mm diameter nozzle is between 0.5 mm and 1.5 mm.) These were allowed to settle to the bottom of the cylinder over a period of approximately 5 minutes.

The mains water supply was then turned on, delivering 5 l/min and exiting the nozzle at approximately 20 m/s. The resulting circulatory, turbulent flow path was strong enough to entrain the polystyrene balls and carry them up out of the cylinder.

This test indicated that, during the piercing stage, the flow should be turbulent enough to carry all of the abrasive and cuttings out of the cement cavity. This assumes that the cutting fluid has been designed so that entrained particles are effectively buoyant within the fluid (i.e. the settling velocity of the particles in the cutting fluid is very slow).

There is a danger that the circulatory motion will be so strong the cement plug will be eroded. Provided the cutting tool is held firmly in place, at a position away from the circulatory flows, this should not be a problem as the section would have to be recemented after the window has been cut (Section 5.5.2).

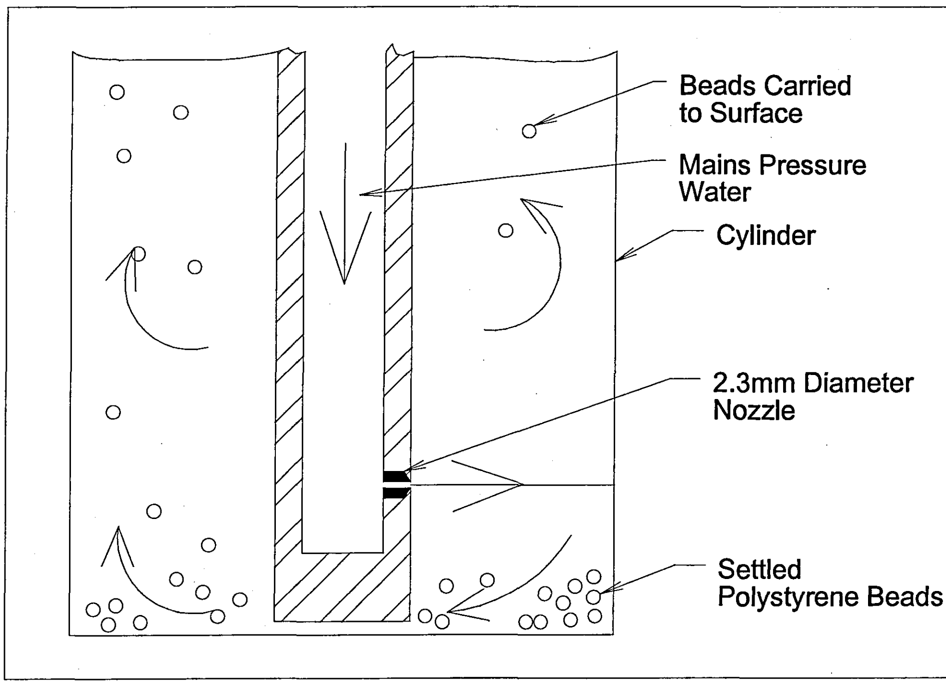


FIGURE 5.5.3 SKETCH TO SHOW THE HOLE CLEANING ARRANGEMENT

5.6 Summary

This chapter has shown that, in theory, a coiled tubing conveyed abrasive water jet window cutting system is feasible. Suitable cutting fluids can be designed and built to transport the abrasive and cuttings, and a single 3.0 mm diameter nozzle should be used at the highest possible pressure in order to cut the window. After cutting the window, the wash-out cavity must be filled with cement to facilitate re-entry with the drilling assembly.

The manipulator design can now be considered. The next chapter presents a detailed Specification for the manipulator, including a summary of all the important points which have so far been identified, before possible manipulator configurations are presented.

Note that the Specification is written as a stand-alone document and therefore contains the relevant references.

6.0 MANIPULATOR SPECIFICATION

AN ABRASIVE WATER JET WINDOW CUTTING SYSTEM

1. Window Location

a) Depth of Operation

From a review of existing oil wells, an operating depth of 3000 m or over would cover the majority of wells for which this application may be applicable.

The maximum depth of operation of an abrasive water jet will depend on the deterioration of the cutting performance with depth and the maximum surface pressure that can be used.

b) Angle of the well.

The window may have to be cut in a well from the vertical position to the horizontal and either on the high or low side of the casing.

The manipulator may have to pass around a bend in the well. The radius of the bend can be split into (Ref 41):

a) long radius 700 - 2000 ft, build rate 2 - 6 °/ 100ft
b) medium radius 200 -700 ft, build rate 8 - 30 °/ 100ft
c) short radius 20 - 40 ft, build rate 1.5 - 3 °/ft.

Long and medium radius wells are the most likely to be encountered, the ability to drill shorter radius wells being a more recent development.

c) Access.

The requirement is for the manipulator assembly to pass down through the production tubing. The size of the production tubing used varies around the world, but generally the smallest production tubing is $2\frac{3}{8}$" and $2\frac{7}{8}$" OD, with ID's varying from 1.600" up to 2.441" (Ref 96).

A No - Go Nipple is usually inserted in the end of the production tubing onto which flow control devices can be supported. They further reduce the ID through which the manipulator must pass.

Typically, for $2\frac{7}{8}$" OD production tubing with 2.441"ID, a 2.205" ID No - Go Nipple will be installed and hence the maximum OD of the manipulator should be 2.18" (Ref 97).

Coronado describes through-tubing workover operations in which 2.13" OD tool, including an inflatable packer, will pass through a minimum restriction of 2.18". The inflatable packer can then be set in 7 - 7.63" OD casing. (Ref 89)

As a first assumption the manipulator must pass through $2\frac{7}{8}$" OD production tubing and must have an OD of less than 2.18". If the basic operating principles for a manipulator can be developed for passing through this size of production tubing, then it can be scaled down for $2\frac{3}{8}$" production tubing afterwards.

2. Preparing the Well

A cement plug must be placed at the required point in the casing. An expandable anchor or inflatable packer must first be set to support the cement while it dries.

A hole must be directionally drilled in the cement plug into which the window cutting tool will be positioned. The maximum size of drill bit that can be used is 2.18" OD. An expandable reamer can then be used to widen the cavity and to dress the top of the cement plug to make re-entry easier and ensure the largest size of manipulator can be used (Ref 98)

3. The Window

a) 2.5 m long (Ref 63).

b) The width of the window depends on the ease with which the window can be re-entered with the directional drilling assembly. It is unlikely that it needs to be more than 1.5 to 2 times the diameter of the drill bit, which corresponds to a width of between 3" and 4". See Section 5.6.1 for corresponding angles of rotation.

c) The window will be cut in casing varying from 5" to 7" OD, although smaller casing is possible. This corresponds to wall thicknesses of between 6 mm and 10.7 mm (Ref 96) although thicker wall sections may be encountered.

d) The aim is to cut the window in approximately 10 hours, which is the time required to cut a window below larger production tubing. The spreadsheet performance calculations show that this is almost possible, depending on the size of the window and its distance from the surface. (Section 5.3)

e) No metal or abrasive must be left in the well.

f) Due to the reduction in cutting performance at depth, the required wall thickness to be cut must be doubled when calculating the nozzle traverse speed.

4. Cutting Strategy

Section 5.2 has shown that cutting the window into pieces is impractical because there is no guarantee:

a) that all the metal could be cut and that it would fall into the well.
b) where the metal would collect; it is likely that some would become jammed between the casing and the manipulator.
c) that the metal could be removed from the well.

Consequently the window must be cut by traversing the nozzle assembly over the complete surface area of the window.

5. Nozzle Size

The largest possible nozzles should be used to minimise the potential of nozzle blockage and to obtain the best cutting performance with stand-off distance (nozzles of at least 2 mm diameter).

6. Pressure

The nozzle pressure and nozzle diameter will be chosen to ensure that the window can be cut within the required time limits.

The pressure losses associated with the flow down through the coiled tubing and back up the coiled tubing - production tubing annulus must be considered. The total surface pressure should be as low as possible to minimise the fatigue of the coiled tubing (and to keep the operators comfortable) and should never exceed the operating pressure limit of the coiled tubing.

7. Abrasive Size and Concentration

The abrasive maximum particle size should be less than half the nozzle diameter. The abrasive concentration used should be no more than 10% by weight to minimise the possibility of blocking the nozzle. (McReynolds recommends that sand concentrations of less than 12% by weight can be pumped out of a well.) (Ref 83)

8. Mud Design

A suitable mud composition must be selected to ensure no abrasive particles settle in the well and to carry both the used abrasive and the cuttings back to the surface. The mud must have good pressure loss and screening properties.

A controllable, nozzle by-pass facility would be beneficial to enable a higher volume of low pressure mud, with no abrasive, to be circulated to assist in cleaning the well if necessary.

Mud conditioning equipment will be required to ensure that all the abrasive particles are removed from the mud, especially any fines.

9. Coiled Tubing

The coiled tubing will be selected to give an adequate pressure limit and flow rates.

10. The Manipulator

The manipulator must be able to:

a) Move the nozzle at a constant speed, as determined by the performance program. The traverse must be smooth, with no jumping to ensure no metal is left uncut.

   Traverse speeds between 10 and 200 mm/min are likely. The actuator must be able to develop the required forces over this speed range.

b) Control the nozzle stand-off to within 5 nozzle diameters (Ref 15).

   Section 5.5.1 explains that a continuously variable nozzle stand-off system may be needed. If so, it must be guaranteed to retract to the initial diameter for removal from the well.

c) Completely isolate the translational and rotational motion, or otherwise have sufficient control and sensors to monitor both motions.

d) Rotate in both directions, through at least an angle of 230°, although ideally through 360° to maximise the flexibility of the system.

   Rotational speeds of between 10 and 200 mm/min are likely. For a maximum nozzle radius of 27.5 mm, this equates to 1 to 20 rpm. The actuator must be able to develop the required torque over this speed range.

e) Withstand the reaction forces, particularly of the deflected jet. Side support for the manipulator might be required, although the cement cavity should take some of the load.

f) Survive the erosion due to the deflected jet. Protection will be required.

g) Ideally, sensors will be required to monitor the cutting procedure. This is to ensure no metal is left uncut and to enable the cutting speed or nozzle pressure drop to be adjusted so that the minimum cement and formation is cut.

By not having free access to both sides of the target, this is very difficult. No methods have so far been found.

h) A method of recording the position of the nozzles with reference to the starting point is required so that cutting can be resumed from approximately the same position in the event of an interruption.

i) Internal flows within the manipulator should be as low as possible to minimise the problem of wear.

j) To minimise coiled tubing fatigue, the coiled tubing must not be moved at the surface while it is under high internal pressure.

11. Environment

a) Operating Temperature.

Standard downhole tools are designed to 175°C. Higher temperature wells are encountered, for which additional modifications may be required.

Note that the typical geothermal gradient is 3.64°C/100 m (Ref 39).

A design temperature of 175°C will therefore be used.

b) Abrasive.

Wells contain several types of particles, all of which are highly abrasive and which are likely to damage seals and any moving surfaces.

Due to the nature of abrasive water jet cutting, the speed of these particles in the vicinity of the manipulator will often be high enough to cause local erosion of the manipulator surface.

The particles will consist of:

Sand abrasive.	These particles will be less than half the diameter of the nozzle (< 1.5 mm) and are accelerated through the nozzle to cut the casing.
Metal Particles.	These are generated by the impact of the abrasive particles and are of a similar size.
Cement Particles.	These are generated by the remaining energy of the abrasive jet after it has pierced the casing. The cement is pumped into the well as a slurry and contains no aggregate. They should therefore be of a similar size to the sand abrasive.
Formation Particles.	The size of these particles will depend on the formation composition and the quantity will depend on the remaining power in the jet after it has cut the casing and cement. Ideally, sufficient monitoring of the cutting performance will be possible to minimise the formation damage.

All of these particles need to be removed from the well, either hydraulically carrying them out or by some other means. There is a possibility that the generation of fines could be a problem if they settle in the blow out preventer (BOP) (Ref 63).

Note:

The mass of sand abrasive that is required is in the order of 10,000 kg, which is not outrageous compared to the quantity used for hydraulic fracturing jobs.

Meese's quotes that typical hydraulic fracturing jobs can require anything from 4500 to 90,000 kg of sand proppant. (Ref 100). He explains that typical proppant sizes are 20/40 and 40/60 mesh (0.85 - 0.425 mm nominal sieve opening and 0.425 - 0.25 mm respectively) If the proppant sand is particularly hard and angular it would be suitable for abrasive water jet cutting, although these size ranges are a little small for a 3 mm diameter nozzle. Ideally particles of between 0.5 and 1.0 mm in diameter should be used.

The volume of metal that has to be removed for a window in 5" casing is approximately $3.175 * 10^{-3}$ m^3, which is approximately 25 kg.

c) Corrosive Fluids.

Materials need to be selected to cope with the general corrosive nature of the environment.

For certain wells hydrogen sulphide is present and materials must be selected to minimise the problem of H_2S attack. This is a secondary consideration at present.

12. Minimum Running Time

The window should be cut in 10 hours. But for larger and thicker casing this could be considerably longer. Typically, the life expectancy of a 2.0 mm or larger nozzle is 20 hours running time, after which the nozzle will have grown too large to operate effectively . Any other consumable on the manipulator must therefore exceed 20 hours continuous running.

13. Maintenance

Consumable items such as the nozzle must be easily replaced under "site" conditions. Similarly routine inspections and maintenance at the site must be possible after each operation.

14. Breakdown or Interruption in Cutting

In the event of interruption of cutting it must be possible to relocate the approximate position of the last cut so that the window can be completed. This could arise from many events including a loss of power or a blocked nozzle. In the case of a blocked nozzle, the system needs to be withdrawn, repaired and then correctly relocated in the well.

15. Failure

In the case of failure, sufficient protection must be given to ensure all manipulator components can be recovered from the well. Emergency disconnects and mating ends for fish must be included. A method of depressurising the coiled tubing must also be considered to facilitate removal from the well.

16. Alternative Window Cutting Tools

No information has been found on a window cutting tool suitable for passing down through $2\frac{7}{8}$" production tubing. However there is no reason why existing through-tubing technology cannot be scaled down, even though, at such a small size, controlling the weight on the mill bit would be very difficult.

17. Cost

The cost of an AWJ window cutting system can be broken into consumable costs and hardware costs.

Consumables

Manipulator consumables: Nozzles, seals, wear plates etc.

Abrasive: Maximum flow rate is approximately 100 l/min.
10% by weight abrasive - 10 kg/min abrasive.
10 hour continuous operation - 10,000 kg abrasive.

Typical cost of abrasive: £50 to £400 per ton.

Flow Additives: to increase viscosity and weight.

Wylie (Ref 101) identified abrasive and fluid as the most expensive costs.

Hardware Cost

Manipulator.

Abrasive metering system (DIAJET), high pressure pumps and associated components.

Coiled tubing - this will depend on the fatigue life for abrasive water jet cutting compared to the fatigue life for conventional milling.

Wireline (if required). A wireline injector may also be required.

18. Standards for Downhole Tools

For high pressure equipment, BS5500, the Pressure Vessel Design Code, might apply.

19. Standards for Operation of High Pressure Equipment: Safety

There are no Health and Safety Executive standards for high pressure offshore equipment. The operating procedures defined in the High Pressure Safety Code published by the High Pressure Technology Association will probably apply.

20. Manufacturing Strategy

The manipulator must be easy to assemble and maintain.

21. Weight

Limited by the rating of the coiled tubing.

22. Size

The length of the manipulator will be limited by the arrangement of the coiled tubing injector. The length of the coiled tubing deployed tool is limited to the height of the lubricator being used (up to 50 ft).

Thomeer explained that it is not unusual for a string of logging tools, perforating tools and/or selective treatment tools to be over 30 ft in length (Ref 102).

7.0 GENERAL MANIPULATOR DESIGN

This chapter discusses the main possibilities for translating and rotating the nozzles, presents a potential manipulator concept design and identifies the most critical functions. Results from test rigs built to investigate these findings are also presented. With this information, a possible manipulator design can be finalised and the detailed design can be considered (this is covered in the next chapter).

7.1 Direction of Motion

The Specification explains that to cut the window the nozzle must be translated and rotated in a zig-zag path. These motions should be achievable independently to simplify the process of monitoring the position of the nozzle. Each motion is therefore addressed separately below. The exact requirements of the translational and rotational control depends on the cutting strategy. That is, whether all of the cutting is undertaken while the nozzle is translated, rotated or a combination of the two. The final manipulator design will play a part in determining this strategy.

7.2 Translation

The Specification details that the nozzles must be translated over a distance of up to 2.5 m, either:

a) at a controlled, smooth speed of between 10 and 200 mm/min.
b) in equal steps, where each step is equal to or less than the nozzle diameter.

The Specification explains that, in order to minimise fatigue, the coiled tubing (CT) cannot be moved from the surface. Moving from the surface alone would not be a viable solution anyway because, due to the friction between the tubing and the production tubing, there would be no guarantee that a small displacement at the surface would give any motion of the buttonhole assembly (BHA). In fact, the BHA tends to jump as the friction is overcome. This is called the Slip-Stick effect.

Similarly, although prototype downhole tractors are available for pulling tools downhole, the Slip-Stick effect will still make the required accuracy in the movement almost impossible. The only alternative is to move the nozzle assembly locally.

To achieve this, some form of telescopic section must be incorporated into the manipulator (see Figure 7.2.1). Basic telescopic sections have been used in CT tools in order to generate large downhole forces (Ref. 99) but no accurate speed control of the stroke is possible. Some form of actuation device is therefore required to extend the telescope at the required accuracy.

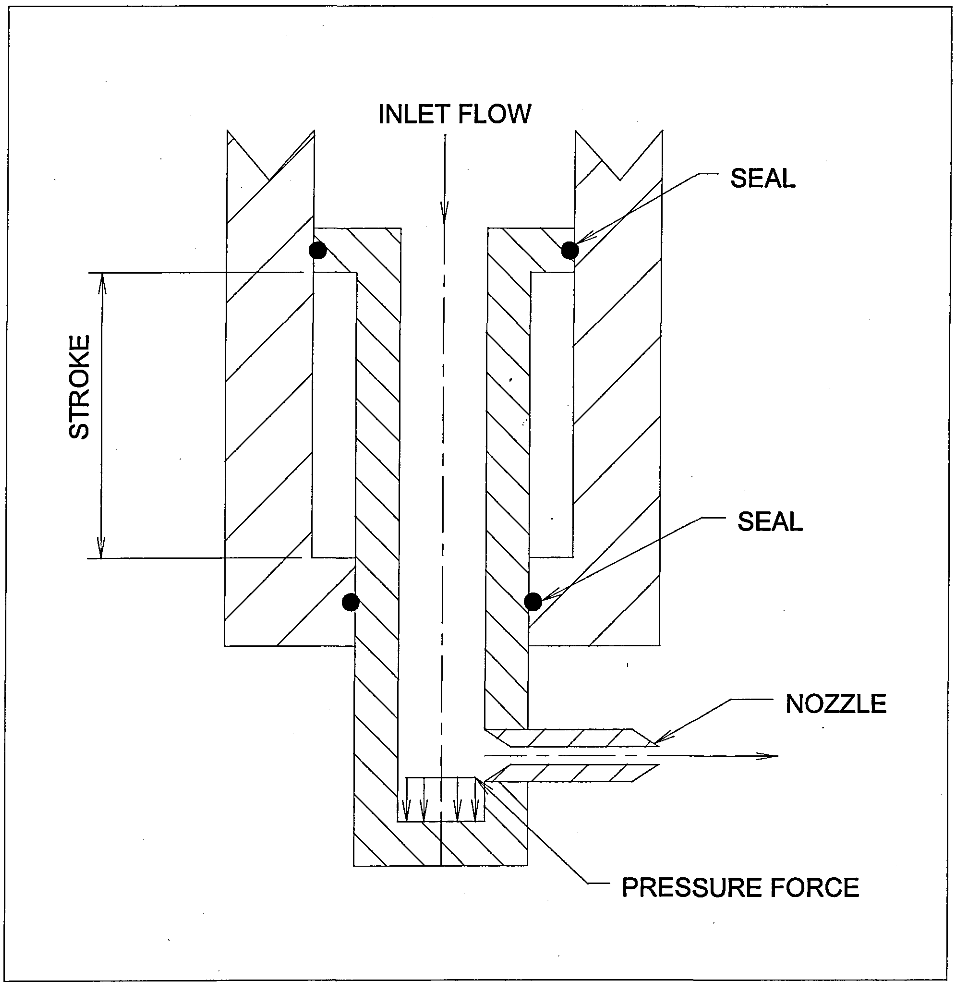


FIGURE 7.2.1 SIMPLE TELESCOPIC SECTION

If the nozzle is attached directly to the telescopic section as shown in Figure 7.2.1, the actuator must be able to apply sufficient force to overcome the large pressure force at the end of the lower telescopic section, otherwise the telescope will just fully extend when pressure is applied.

For example, if the OD of the manipulator has to be less than 55 mm (2.18") then the ID of the inner telescope is likely to be approximately 20 mm. If the nozzle pressure is in the order of 350 bar then the end force on the lower end of the telescope is approximately 11000 N. This is an enormous load to react. Consequently the telescopic section must be pressure balanced. Then there is no tendency for the telescopic section to move when the system is pressurised. A pressure balanced design is shown in Figure 7.2.2.

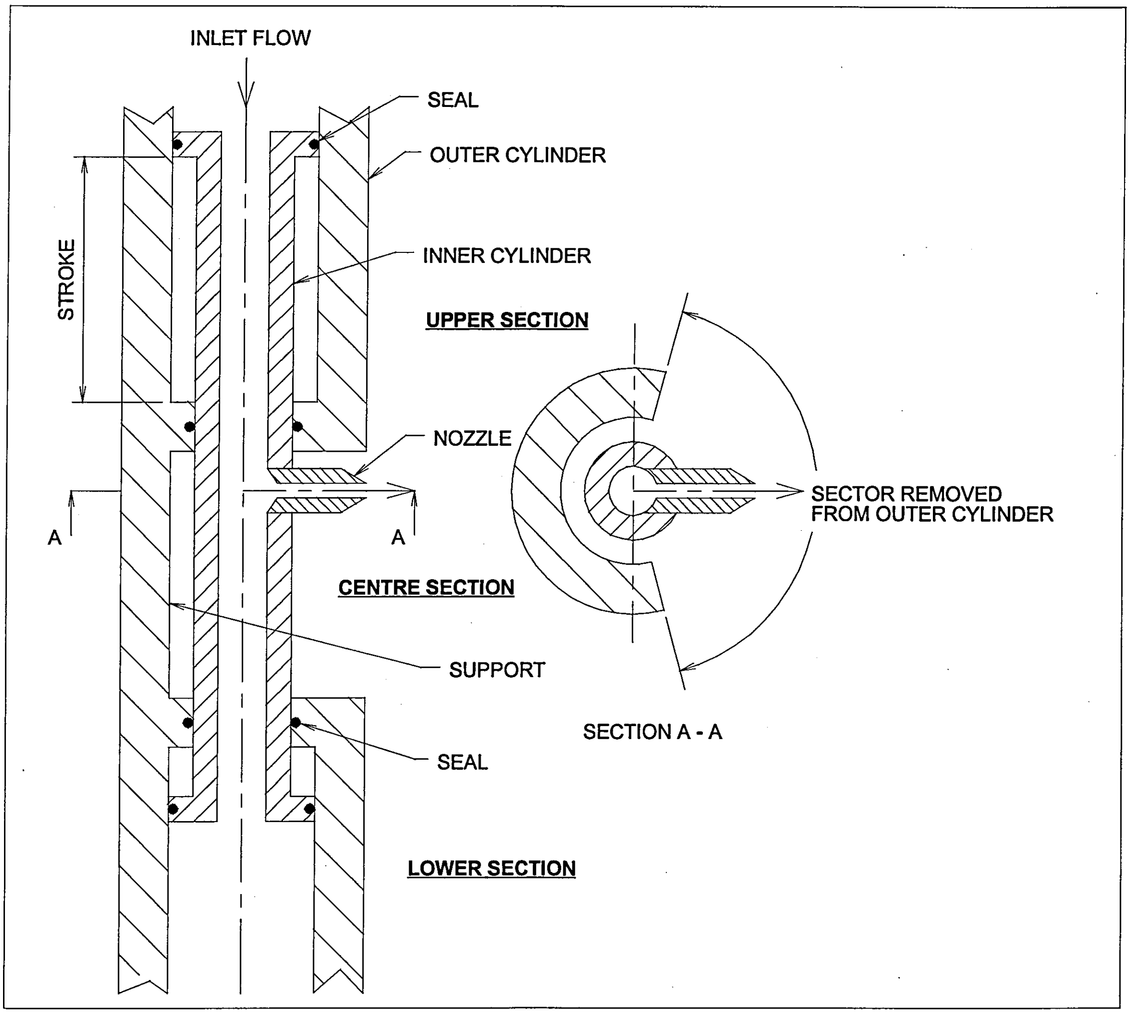


FIGURE 7.2.2 PRESSURE BALANCED TELESCOPIC SECTION

An actuator is still required to accurately move the nozzle and inner cylinder. **However, it now only has to generate sufficient load to overcome the frictional forces between the seals**. There are two methods of actuation:

1) Hydraulically -

Either a hydraulic ram could be used to move the inner cylinder, or the sealed cavities between the outer and inner cylinder could be differentially pressurised. In both cases a pump and valving system would be required to generate the hydraulic flow and so a method of powering the pump is required. The main problems with using hydraulics include:

- accounting for the compressibility of the hydraulic fluid if high pressures are required to overcome the friction forces, this makes control critical.

- incorporating miniature valves and pumps into the confined space of the manipulator.

- powering the pump. The hydraulic power of the abrasive flow could be used, but the turbine would wear rapidly, or the pump could be powered electrically, in which case a wireline cable must be installed in the CT. Alternatively a metal hose could be installed in the CT to convey the necessary hydraulic flow from the surface.

2) Mechanically -

A lead screw is commonly used on machine tools to give accurate motion control. This could easily be coupled to the inner cylinder and driven by an electric motor. Again, a wireline cable would have to be used. (Note that miniature batteries are not yet powerful enough to provide the required power and life).

Suitable brushless permanent magnet motors which are used regularly inside oil wells are available from Kollmorgan Hightech Limited (see Appendix G)

Of the two options, a lead screw is far simpler to incorporate into the confined space and gives a much more definite level of accuracy.

The stroke of the telescope is important. This is determined by manufacturing limitations, the stability of the support linking the upper half of the manipulator to the lower half of the manipulator, and the critical buckling load of the lead screw. For a 2.5 m stroke, the length of the manipulator would be over 10 m (2.5 m upper section, 2.5 m centre section, 5 m lower section, which consists of 2.5 m for the inner cylinder and 2.5 m for the lead screw). The compressive loads acting on the support, shown in Figure 7.2.2, over this distance will cause it to deflect and even buckle. The inner cylinder will then start to carry the deflected loads, increasing the required actuation forces, until it finally locks up.

A stroke of 2.5 m is therefore not feasible, but there is no reason why a shorter stroke is not possible. A second method of translating the nozzle would then be required, but this would need to be achieved with much less accuracy.

The easiest method to move the nozzle assembly over a set distance (speed is of no importance) is to use a system of shear pins (Figure 7.2.3), with each set of shear pins being separated by a distance of less than the stroke. The coiled tubing is then reeled in to shear the pin and move the manipulator to the next pin position. To ensure that the coiled tubing is not moved under high internal pressure, the internal pressure in the coiled tubing must be reduced to below 2500 psi. Also, because the pressure force can no longer be transferred to the locating packer, tension must be maintained in the coiled tubing to react against the pressure force and keep the manipulator in the correct place.

The sequence of operation would be:

a) Insert the manipulator in the well and locate on the packer in the cement plug.

b) Reel in the coiled tubing until the load rises to indicate the coiled tubing has locked into place and that the first shear pin has been reached.

c) Pressurise the coiled tubing with plain fluid, tension being applied to the coiled tubing to maintain the position of the manipulator.

d) Start pumping abrasive and once sufficient time has been given to allow it to reach the nozzle, start cutting over the short stroke.

e) When close to the end of the stroke, stop the abrasive flow and flush the line with plain fluid.

f) When cutting is finished and all the abrasive flushed from the coiled tubing, depressurise at the surface to below 5000 psi.

g) Reel in the coiled tubing to break the first shear pin. The tension in the tubing will then drop until the coiled tubing reaches the next shear pin. Stop reeling, repressurise, start the abrasive flow and start cutting the next section.

Shear pin assemblies are incorporated regularly in downhole tools. To prevent the possibility of the tool suddenly jumping and shearing 2 or 3 sets of pins in the same pull, the pins should be graduated in strength. The exact design of the shear pin assembly should be carefully considered to ensure the broken pins cannot become trapped in the mechanism, preventing motion. The detailed design will not be considered as it is conventional oil industry technology.

Note that References 60 and 65 detail that shear pins are generally rated to fail at 40,000 to 60,000 N, although as high as 100,000 N has been used.

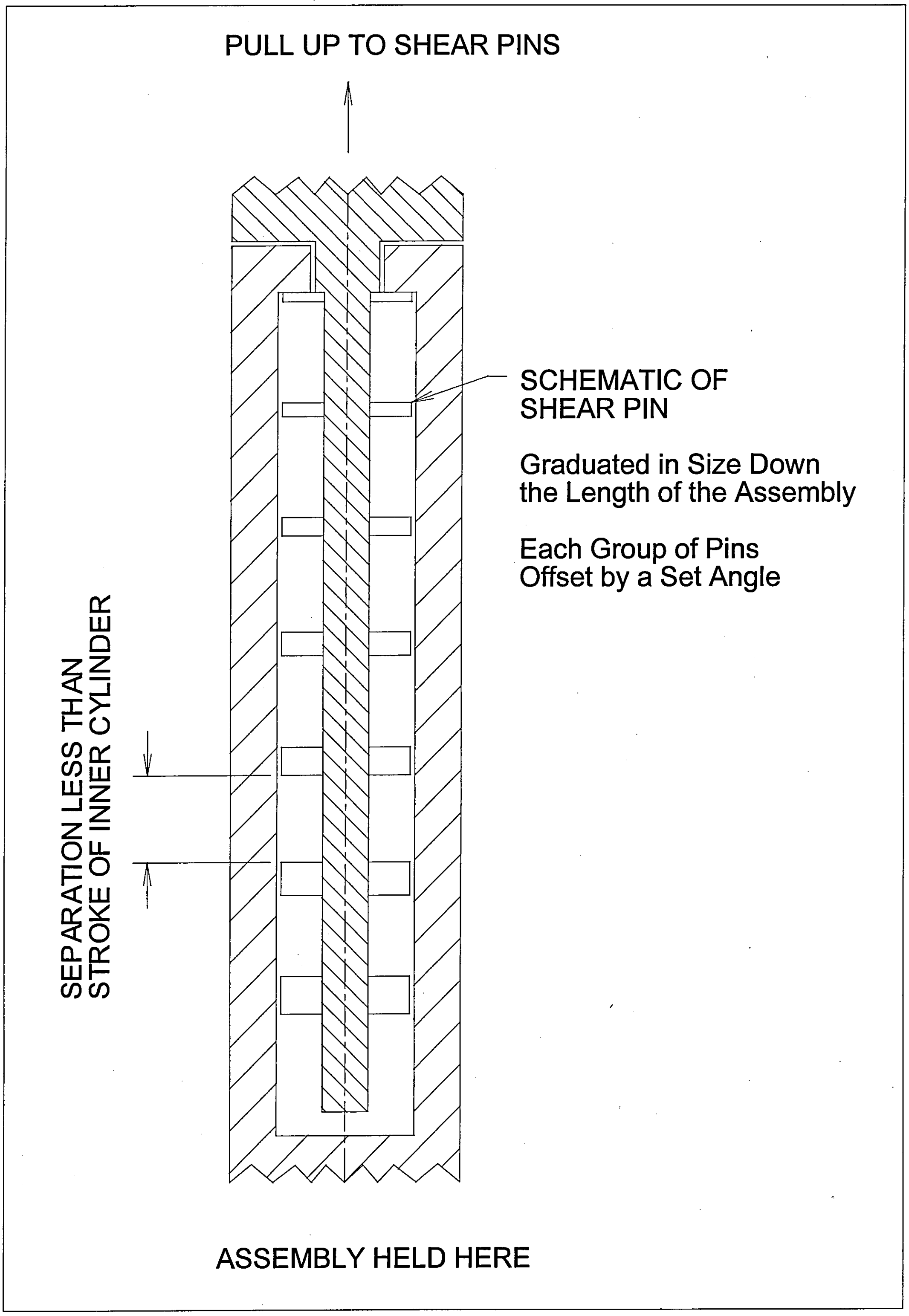


FIGURE 7.2.3 BASIC SHEAR PIN ARRANGEMENT

7.3 Rotation

The Specification details that the nozzle must be rotated either:

a) through a small step equal to the arc length corresponding to the diameter of the nozzle.

b) through an angle of at least 230°, depending on the width of the window, at a rotational speed of between 10 and 200 mm/min (1 to 20 rpm). This rotation must be smooth with no jumps. Ideally, to reduce the criticality of inserting the manipulator correctly orientated in the well, 360° rotation should be possible. The operator must be able to measure the orientation of the nozzle.

If the basic telescopic arrangement is used as shown in Figure 7.2.2, then there are two basic options for rotating the nozzle:

1) Rotate the complete outer cylinder and inner cylinder assembly together. This would require incorporating a rotating union (swivel) at the top and bottom of the outer cylinder, allowing full 360° rotation. Although suitable swivels are standard components in the oil industry, the frictional forces to move them can be high and to satisfy the maximum OD requirement, their internal bores tend to be very small (Ref 103). Finally, incorporating both the rotary and linear actuators into this could be difficult.

2) Rotate the inner cylinder only. For this there are two options:

a) have the outer cylinder support as shown in Figure 7.2.2, with a section of the outer cylinder removed. This method gives a limited angle of nozzle rotation which means the manipulator must be orientated exactly right in the well when it is deployed. To achieve 230° coverage, insufficient support will be provided by the remaining outer cylinder to maintain the correct alignment of the upper and lower sections. Additional support will therefore be needed.

b) to achieve 360° rotation of the inner section, the complete outer cylinder in the centre section would have to be removed. An alternative method of supporting the upper and lower sections is then required. This could be just a simple column with a slot along one side (see Figure 7.3.1) along which the nozzle can be traversed. Bearings can be incorporated at either end of the support column, so that when the inner cylinder is rotated the support column rotates with it.

Option 2b is possibly the best, but it requires that the seals are suitable for both rotary and linear motion.

Method of Actuation

The same basic methods of actuation apply to rotary motion as to translational motion. Consequently, if an electric motor can be used to rotate a lead screw there is no reason why a similar motor could not be coupled directly to the inner cylinder. The brushless permanent magnet motors described earlier include a gear box to generate high torques and encoders to give excellent speed and position control.

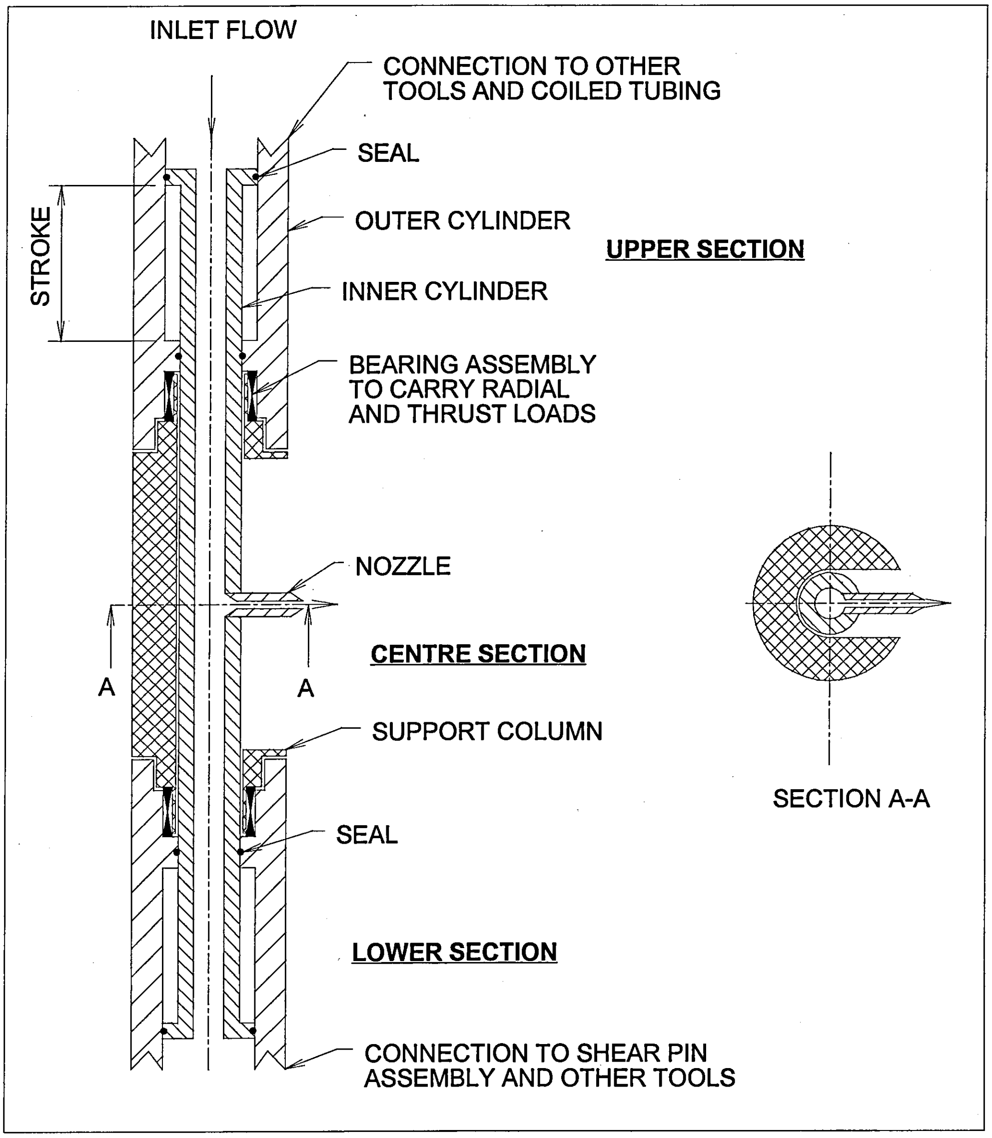


FIGURE 7.3.1 PRESSURE BALANCED TELESCOPIC SECTION WITH SUPPORT COLUMN

7.4 Overall Design

The basic conceptual design for the telescopic section and actuators is shown in Figure 7.4.1. The nozzle must be translated and rotated in a zig-zag path to cut the window. The window will be cut while the nozzle is rotated. At the end of the rotation, the nozzle is translated a distance equal to the nozzle diameter.

The shear pin assembly would be mounted below the telescopic section and actuators. The bottom section of the shear pin assembly fixes into the packer placed in the cement plug. Two permanent magnet motors are required to provide the necessary rotation and translation and these are mounted in a motor support housing. This housing has two anti-rotation pins which locate in slots in the outer cylinder. These pins are essential for the working of the system:

- If the inner cylinder has to be translated, but the force required to translate is greater than that required to rotate, then instead of turning the lead screw, the motor and inner cylinder will rotate. The anti-rotation pins prevent this from happening.

- Similarly, if the inner cylinder has to be rotated, but the force required to rotate is greater than that required to translate then instead of turning the inner cylinder, the motor and lead screw will turn. The anti-rotation pins again prevent this from happening.

The position of these anti-rotation slots in the outer cylinder is critical. They must start below the bottom point of the stroke of the inner cylinder, otherwise the seals in the inner cylinder will not seal correctly.

Note that the motor assembly is positioned below the inner cylinder so that it is not in the abrasive flow. If required, for additional protection, a filter can be incorporated into the lower part of the inner cylinder.

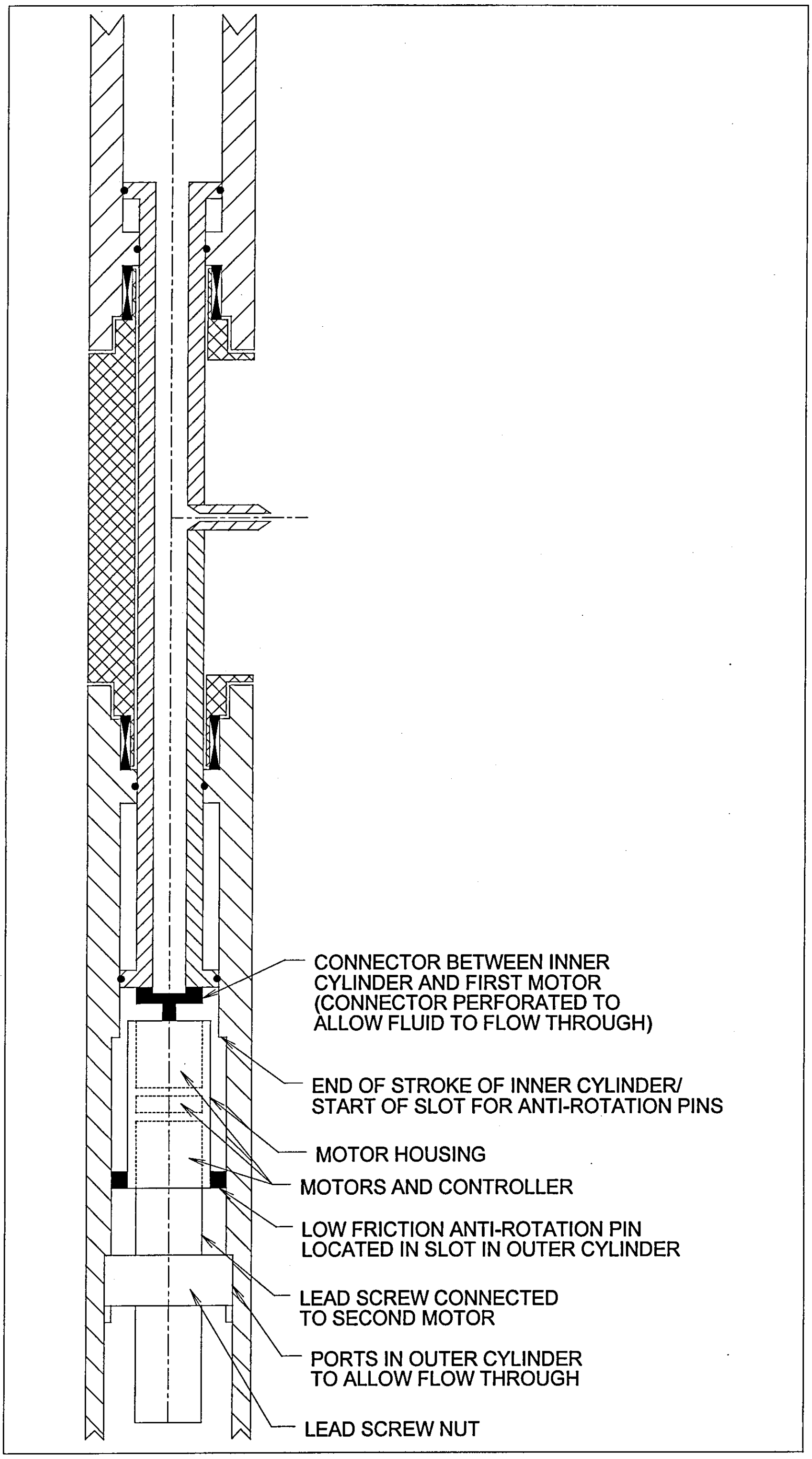


FIGURE 7.4.1 SCHEMATIC OF MANIPULATOR INCORPORATING MOTORS AND PRESSURE BALANCED TELESCOPE

7.5 General Concerns

There are many general concerns with this design which must be addressed before detailed stress calculations can be conducted. These are:

- the wear of the outer surface of the inner cylinder due to the reflected abrasive jet.
- the entrapment of abrasive in the inner cylinder/support column cavity.
- the selection of suitable seals and the frictional forces required to move them under the operating conditions.
- the construction of the manipulator for assembly and easy maintenance.
- the nozzle reaction force.
- the nozzle design and the need to continually know the orientation of the nozzle.
- the flow velocity through the inner cylinder, particularly with a cable installed.
- a method of flushing the coiled tubing clear of abrasive in the event of a blocked nozzle.

Each of these points is covered below.

7.6 Wear of the Outer Surface of the Inner Cylinder Due to the Reflected Abrasive Jet

Section 5.5.4 explains the basic cutting process and how the abrasive jet deflects back to the manipulator. The deflected jet will erode an area around the nozzle. This area needs to be well protected. It is also essential that the outer surface of the inner cylinder is not eroded by the jet; the outer surface is a sealing surface which must be machined to a specified surface finish if the seals are to seal properly.

Ceramic plates positioned around the nozzle will give adequate protection but the extent of the erosion zone due to the deflected jet needs to be determined. A model cutting head has therefore been designed and built to investigate the basic process of cutting a window (see Section 7.8)

7.7 The Entrapment of Abrasive in the Inner Cylinder/Support Column Cavity

Abrasive will be continually moving around the manipulator. It is possible that abrasive could collect in the cavity between the Inner Cylinder and the Support Column. This will increase the likelihood of abrasive becoming trapped in the seals and bearings, consequently damaging them, and in the worst case the concentration of abrasive might be so high the inner cylinder locks up entirely.

A model of the cutting head has been designed to incorporate a cover to simulate the support column in order to investigate more clearly where abrasive might collect.

7.8 Model Cutting Head

7.8.1 Basic Design

Figure 7.8.1 shows the basic arrangement of the Cutting Head and the manufacturing drawings are in Appendix D. The principle of the design is a simple stainless steel tube with a high pressure fitting at the top, rated to 700 bar. A 0.5 mm diameter nozzle is held in a tapered nozzle holder, which screws into the side of the steel tube, and a wear plate assembly protects the nozzle from erosion by the deflected jet. A mild steel metal tube with a 100° sector removed from one side, slides over the Cutting Head so that the nozzle and wear plate project out of the assembly. This represents the hollow support column shown in Figure 7.3.1. 'O' rings are used to form a light contact seal between the housing tube and the Cutting Head so that if abrasive is present in the region of the 'O' ring, it should become trapped and will not fill the cavity below the Cutting Head.

The Cutting Head and housing are installed inside two metal tubes which have been cemented together to simulate oil well casing. The complete assembly is then submerged in water and the Cutting Head is connected to a DIAJET set to discharge 15% by weight of olivine. By operating at approximately 300 bar and by slowly rotating the Cutting Head, crude slots can be cut into the inner casing.

Note that the outer casing is a protective cover which prevents the jet damaging the tank. The end plug forms a water-tight seal with the inner casing, which ensures the nozzle stays submerged.

To obtain the best simulation of the real problem, the tubing is sized as close as possible to the actual dimensions quoted in the specification.

The maximum nozzle stand-off is approximately 5 mm which is much greater than the stipulated 5 nozzle diameters for submerged cutting.

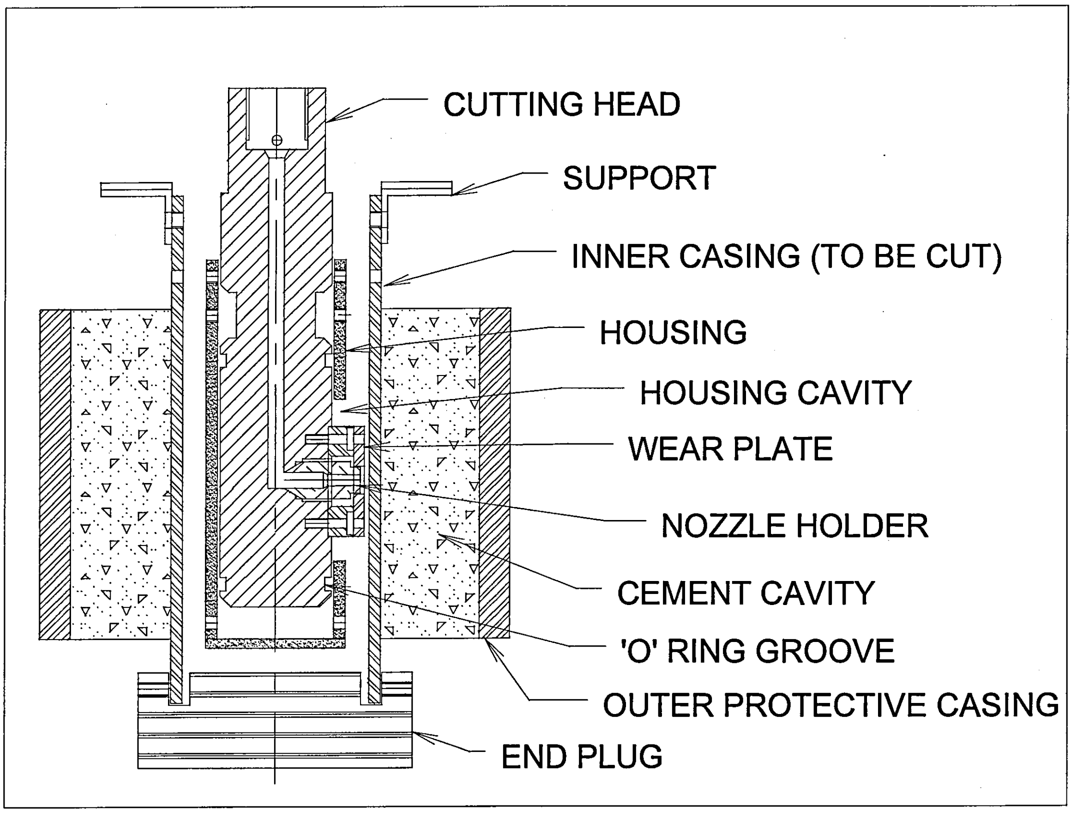


FIGURE 7.8.1 SIMPLE CUTTING HEAD MODEL

7.8.2 Results

The following series of tests, with the main results and observations, were conducted:

Test 1) Run at 100 bar with the nozzle submerged and stationary.

When the abrasive was turned on, it could be clearly seen flowing out of the top of the inner casing.

When the DIAJET was turned off and the Cutting Head removed, some abrasive was found to have settled to the bottom of the housing cavity - the 'O' ring was too slack to seal completely. No abrasive was found in the bottom of the Cutting Head.

Abrasive also settled above the top 'O' ring and small amounts had stuck to the inside of the housing, in the housing/Cutting Head cavity.

By removing the Cutting Head and replacing it with an ordinary mains hose, which delivered water at 5 l/min, all of the abrasive was washed up out of the inner casing.

Some abrasive had also settled to the bottom of the casing, although most had been flushed out and was lying on the top of the cement.

There was no sign of wear on the wear plate or exposed part of the nozzle holder.

Test 2) The assembly was cleaned and rebuilt. By accident, while trying to repeat the above test, the end plug at the bottom of the inner casing fell off. The nozzle was then operating in air and wear was evident up to a radius of 16 mm from the centre of the nozzle.

Test 3) The rig was correctly assembled and tested at 300 bar. At this pressure, the velocity of the plain water jet was sufficient to remove all of the water out of the inner casing. The nozzle was effectively operating in air again and the abrasive was not turned on.

Test 4) The abrasive was turned on at 100 bar and then the nozzle pressure was increased to 300 bar. The nozzle was held stationary and after a few seconds, the jet pierced the inner casing and cut a cavity in the cement. After 20 seconds, the cavity extended to the top of the cement and the jet escaped through the hole in the cement. Some wear was evident on the nozzle holder and wear plate (up to a radius of 12 mm from the centre of the nozzle). There was no evidence of wear of the housing or Cutting Head. There was no sign of large pieces of concrete, except where the surface concrete had been blown out rather than cut.

Note that the wear on the wear plate tended to be on one side of the plate. The wear was visible as a light polishing of the surface. This is in contrast to the more significant wear on the nozzle holder.

Test 5) Test 4 was repeated at a different position in the casing. This time the Cutting Head was slowly rotated, and a short slot was cut in the casing. Again, the jet eventually escaped out of the top of the cement.

Test 6) To investigate the possibility of abrasive getting into the cavities between the housing and the Cutting Head, the cavities were filled with silicon sealant. Test 1 was immediately repeated so that the sealant remained viscous.

Although this was a very crude attempt at trying to prevent the entrapment of abrasive, no abrasive was found below the Cutting Head and there was no evidence of the sealant being washed away.

Test 7) After completing these tests, the Cutting Head was mounted on the X-Y table and a cut made in 25 mm thick stainless steel 304L at 690 bar and 6 mm stand-off. This was conducted to investigate the effect of a 90° degree bend just upstream of the nozzle. The test was repeated with a standard nozzle without the bend.

The maximum parting speed (50 mm/min) was the same in both cases. This is not surprising considering that the minimum distance between the nozzle inlet and the bend was 13 mm (26 nozzle diameters); a sufficient distance for the turbulence generated by the bend to start to die.

7.8.3 Conclusions of Cutting Head Trials

The tests showed that:

1) Not all of the abrasive was washed out of the inner cylinder. Some settled below the nozzle. By increasing the water flow rate and directing it at the abrasive, all of the abrasive was washed out.

   The window cutting manipulator needs to include a bypass valve so that a larger volume of abrasive-free fluid can be pumped downhole to flush away abrasive. The flushing port ideally needs to be directed towards the settled abrasive.

2) There was some limited erosion of the wear plate and exposed nozzle holder but no erosion of the housing or Cutting Head. The erosion of the exposed 17-4 PH stainless steel nozzle holder (approximate hardness 30 RC) was significant compared to the wear plate, which had been heat treated to a minimum surface hardness of 60 Rockwell C to give good wear properties.

   The window cutting manipulator needs to have a sufficiently large wear plate to catch the deflected jet and must cover all of the nozzle holder. The holder may have to be of a hardenable material to improve its wear properties.

3) Abrasive was found to collect in every cavity in the manipulator. This was partially because the 'O' ring seals were not tight enough. However, by filling the cavities with silicon sealant, only minimal quantities of abrasive collected.

   The cavity between the inner cylinder and support housing must be filled with a suitable environmentally friendly grease or an alternative filler. Access ports are needed in the side of the support column to ensure the whole column is filled with grease. The grease must not have a detrimental effect on the bearings at either end of the column. The additional forces required to overcome the viscous forces need to be considered. If possible, a protective cover is required over the slot in the support column to minimise the possibility of the grease being washed away.

4) The nozzle holder screwed into the side of the Cutting Head body and sealed on a 60° cone. Although this was successful, the nozzle holder was too delicate to be reassembled repeatedly. The 60° cone and screw threads would soon deform and no longer seal. (Note that, for these tests, the nozzle holder had to be removed several times to unblock the nozzle)

A greater worry was the potential for the nozzle holder to bind with the Cutting Head body. On several occasions I have experienced galling of stainless steel threaded connections, including the combination of 17-4 PH and 316 used here. A special anti-galling lubricant was used to minimise the likelihood of galling. These lubricants are expensive, but are essential when the alternative is the loss of the Cutting Head.

For the final manipulator, the nozzle will be removed at the end of each operation, either to unblock it or to replace it. An alternative method of holding the nozzle in place therefore needs to be considered to take account of the need to regularly replace it. If the same nozzle design is used, it must be more robust than the design used here.

5) The nozzle design has already been identified as an important consideration and will be discussed later. Other tests are needed to investigate the cutting performance when a large nozzle (> 2.0 mm in diameter) is positioned directly after a 90° bend. A reduction in cutting performance is expected because of the worse inlet conditions compared to a standard nozzle arrangement. If there is a reduction in cutting performance, this needs to be included in the cutting model.

Where possible these recommendations are included in the detailed design section.

7.9 Seal Selection

The seal selection is critical to the success of this design.

Firstly, the seals must be able to allow both rotational and translational motion in an abrasive environment at temperatures up to 175°C. They must satisfy the target running period of at least 20 hours.

Secondly, the frictional forces between the seals and sealing surfaces must be minimal. There is only limited space available to house the electric motors and consequently the actuation forces they can generate are limited. If the friction forces are too high then the proposed design will never be feasible.

BHR Group have a close working relationship with James Walker Limited, a seal manufacturer. I therefore approached them for recommendations for the best seal arrangement. Based on my pressure balanced telescope design, they suggested the following (see Figure 7.9.1).

1) A wiper made of a material called Zymax™. This should clear abrasive from the sealing surface and consequently protect the main seal. The wiper must contain a couple of very small holes (much smaller than the size of the abrasive particles) to ensure there is no pressure differential across it.

2) A composite main seal which consists of Zymax and Elast-O-lion 175.

3) A split bearing, again made from Zymax, to support the inner cylinder.

They also recommended an 'O' ring of Elast-O-lion 175 which would be suitable for sealing the screw threads. Note that depending on the position of the 'O' ring, an anti-extrusion ring may also be required.

The Zymax material is particularly suited for use in hot abrasive environments. Also, a representative of James Walker Limited believed that, because of the very slow actuation speeds required, the seals should last the minimum 20 hours of rotation and translation, although they had no relevant test data.

They could not specify the frictional forces required to move the seals.

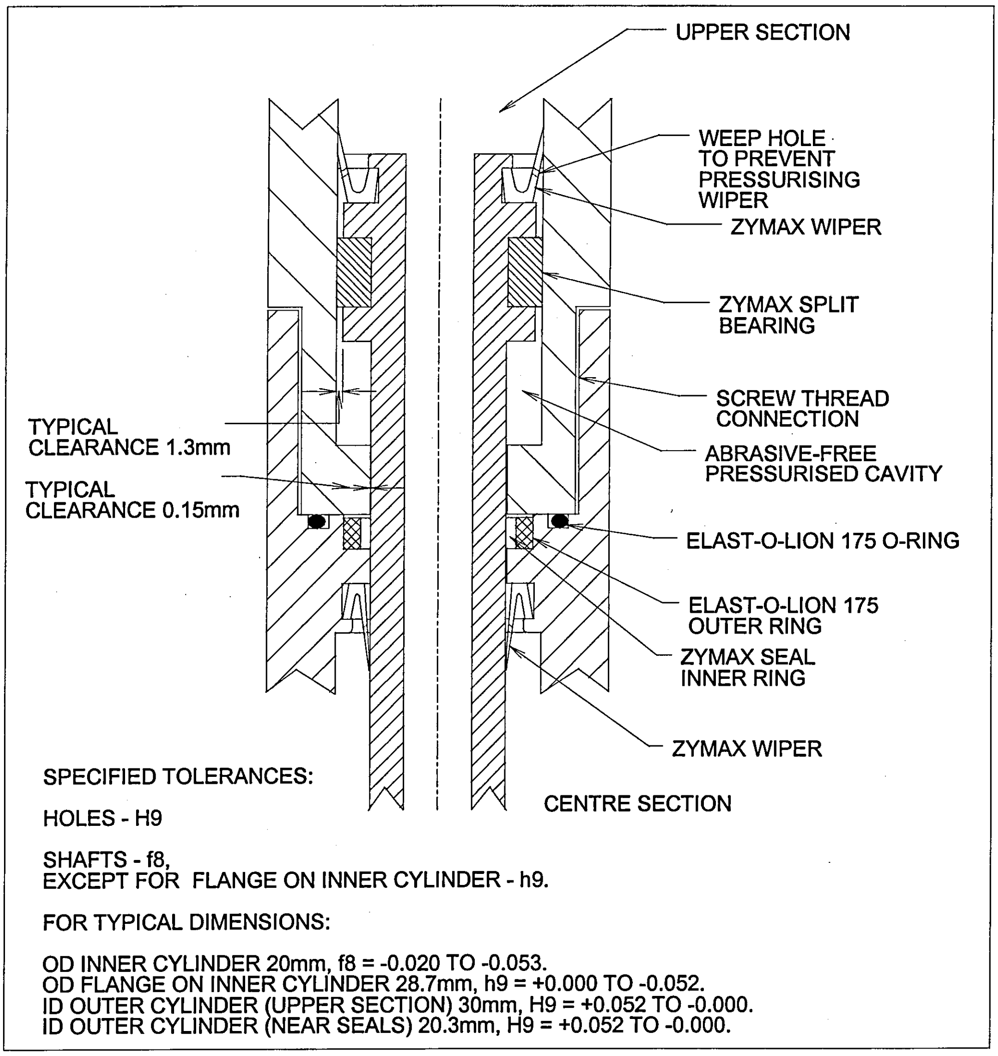


FIGURE 7.9.1 PROPOSED POSITION OF SEALS, WIPERS AND BEARINGS IN UPPER HALF OF PRESSURE BALANCED TELESCOPIC SECTION

Note that the same seal arrangement would be used in the lower half of the pressure balanced telescopic section.

Note that there is an abrasive-free pressurised cavity. Cutting fluid can pass between the wiper seal, bearing and housing, abrasive cannot.

A brief review of piston seal testing conducted by the Seals Department of BHR Group revealed the following points relating to the frictional force:

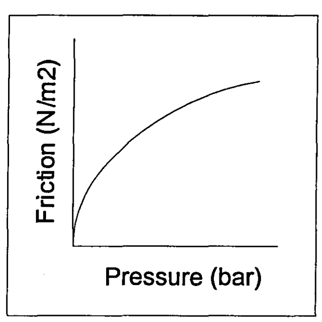
1) For an elastomeric seal, sealing on a 50 mm nominal diameter shaft at a maximum pressure differential of 300 bar, the maximum frictional force (break-out friction) was measured at 4000 N (Ref 104). (This was for translation only.)

   Note that the diameter of the seals used for the manipulator will be in the order of 20 mm. The manipulator frictional forces should therefore be considerably lower.

   There are 2 seals in the pressure balanced telescope, 4 wipers and 2 bearings, as shown in Figure 7.9.1. The break-out friction associated with the wipers and bearings is unknown, but will be considerably less than that of the seals. As a first approximation, to account for these unknown frictional forces, assume a 100% safety factor on the break-out friction of the seals; i.e. assume there are effectively 4 seals. Then the maximum frictional force to be overcome is 16000 N.

2) In general the measured frictional force tends to increase more slowly as the pressure differential across the seal increases (Graph 7.9.1).

GRAPH 7.9.1 GENERAL SEAL FRICTION AGAINST PRESSURE CURVE FOR A RECIPROCATING SEAL



3) For rotary seals, the seal friction is approximately $= Dwf$

D	-	diameter of sealing surface
w	-	contact width
f	-	specific friction

   However, the value of f for these seals is unknown. As a first approximation, I was recommended to try 100 lbf/in^2 of seal (Ref 105).

Assuming a seal having dimensions $w = 3.2$, $d = 20$ mm

$$\text{the frictional force} = \pi\left(\frac{3.2}{25.4}\right)\times\left(\frac{20}{25.4}\right)\times 100 = 31 lbf \equiv 138N$$

The break-out torque is therefore $= fd$
$= 138 \times 20 \times 10^{-3}$
$= 2.8\ Nm$

Again, assuming a 100% safety factor for the seal assembly, the total torque required to rotate the nozzle would be 11.2 Nm.

4) The frictional force can vary by as much as 100% over a short period of time, which means very large factors of safety must be incorporated at the motor selection stage.

Note that these frictional forces are **very** approximate. During the brief review of seal performance, no simple prediction models were found as there are too many variables.

Seal selection for this application is a complex task. For the purposes of this study, only a base value for the frictional force for these particular seals at the specified operating conditions is needed. The only way to obtain this information is to build a seal test rig. If the seals then prove to be unsuitable, alternative seal designs need to be investigated.

The actual manipulator will be exposed to abrasive particles of 1.0 mm diameter or less. Although very fine abrasive particles will be formed after cutting, they should be prevented from reaching the main seals by the protective cover and grease filled cavity. Internally, very few particles will be small enough to pass the wiper clearance. Testing the seals on plain water should therefore give a realistic indication of the break-out forces for the final manipulator.

7.10 The Seal Test Rig

7.10.1 General Design

A seal test rig was designed to operate on water at up to 600 bar and 175°C (see Figure 7.10.1). The seal rig was designed to incorporate the same size of seals as those selected for the basic manipulator, ensuring that the break-out frictional forces should be as close to the actual conditions as possible. Instead of an inner cylinder, the seals sealed on a plain 20 mm diameter shaft.

James Walker Limited generously supplied a set of seals (valued at £500) free of charge. Unlike conventional 'O' rings which can be stretched into position, the main Zymax seals are relatively stiff. Consequently, a split seal housing was required. A split housing design, based on the manipulator design, was used to demonstrate that the housings could be machined accurately enough that there was no leakage and that there was insufficient misalignment to interfere with the shaft.

The split housing consists of a central wiper housing and seal housing connected by an M45 screw thread. An O-ring and back-up ring are incorporated in front of the thread to seal the joint.

Note that a representative of James Walker Limited recommended positioning this O-ring in a groove in the face of the housing (see Figure 7.9.1), to minimise the chance of damaging it. Unfortunately there was insufficient wall thickness to do this and instead the O-ring had to be installed with a anti-extrusion ring around the circumference. This method of sealing has the advantage that it seals more readily, no specified torque needs to be applied to the screw thread to ensure the O-ring is sufficiently compressed. Also the O-ring groove is easier to machine. By greasing the threads thoroughly and assembling the joint carefully, I had no trouble assembling these seals.

For these tests, the wipers had to be mounted internally to the seals. To achieve this the wiper seal was housed in a separate back up ring which also formed part of the main seal groove. The back up ring is held in place by the main housing. A split bearing is also positioned in the seal housing to support the shaft.

Mounting the composite main seal proved to be very difficult. The Zymax was far stiffer than I had realised and the bottom of the seal groove was very inaccessible, being over 40 mm from the top of the housing. A separate tool was really required to compress the outer elastomer around the Zymax seal and to insert it into the groove. An alternative seal housing arrangement should have been used, based on the back up ring design which houses the wiper seal. These assembly problems must be considered in the final manipulator design.

For the seals to seal properly, the shaft must have a minimum hardness of 40 Rockwell C (RC). There is always the danger that abrasive will become embedded in the seal and bearing material, resulting in the scoring of a soft shaft and a reduction in the seal efficiency. Usually a suitable steel shaft would be nickel plated to give a surface hardness of 48 to 52 RC. However, for the real manipulator there is the concern that the nickel plating could peel off, particularly if it was hit by a deflected jet, and that the steel below may corrode.

An alternative solution was to use an austenitic stainless steel shaft and have it specially surface treated, by a process called Kolsterizing, to give a surface hardness of over 60 RC to a depth of 33 μm. This process does not alter the tensile and corrosion properties of the stainless steel and was therefore thought to be a more reliable solution. The only worry was that the hard surface was too thin and that the softer material below would deform under the high seal loads. There was no evidence of this during the test.

All surfaces touching the seals had to be machined to a surface roughness of less than 0.4 μm Ra.

The shaft was connected to a linear actuator, which could generate up to 30,000 N force, with a load cell between, so that the translational friction force could be measured using a voltmeter. The voltmeter measured the peak voltage and therefore the maximum force applied. The maximum force is compressive in one direction and tensile in the other.

The torque required to slowly turn the shaft was measured by connecting a second linear actuator and load cell to one end of a 0.45 m long lever arm and connecting the other end of the lever arm to the shaft. This linear actuator could develop a force of up to 500 N. The zero reading was taken and subtracted from the maximum reading to account for the mass of the lever arm. Again the maximum torque is measured by the load cell as a compressive force in one direction and a tensile force in the other direction.

The linear and rotational tests had to be conducted separately; to rotate the shaft, it first had to be disconnected from the linear actuator. Note that the load cells were calibrated in tension by hanging weights off them.

Initially the aim of the rig was to measure both the frictional forces and the life of the seal. Unfortunately, to conduct realistic life measurements the rig must be operated continuously for 20 hours. The safety monitoring and data logging equipment required to achieve this proved to be far outside my budget (materials, machining and basic measuring equipment cost more than £4000). Consequently only basic friction measurements could be measured.

The stress calculations are discussed in Chapter 8 and the manufacturing drawings are given in Appendix E.

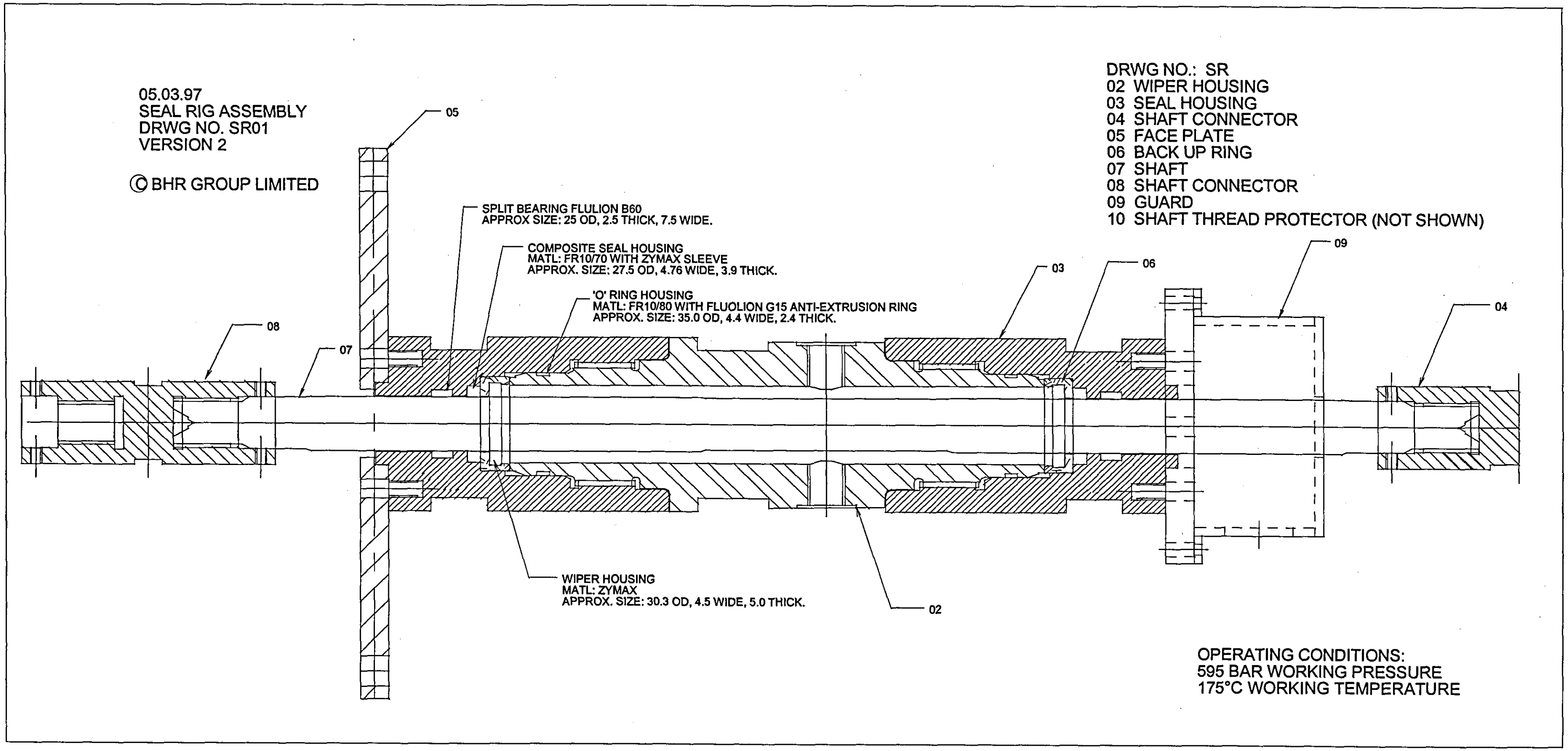


FIGURE 7.10.1 SEAL TEST RIG

7.10.2 Measurements

Linear:

Three proximity switches mounted alongside the linear actuator ensured that the shaft moved in steps of approximately 10 mm. When a marker, moving with the linear actuator, was detected by the proximity switch, the linear motion was stopped. This sequence was controlled by a PLC program. Separate emergency end stops prevented the actuator moving too far.

To check the repeatability of the friction measurements, a total of 8 measurements were taken: if the marker started at proximity switch 1, it was moved to switch 2 and the maximum frictional force measured. This is reading Number 1. Then the marker was moved to switch 3 and again the maximum frictional force measured; reading Number 2. The direction of motion was then reversed to give readings Number 3 and Number 4. The whole sequence was then repeated to give a total of 8 readings.

Rotation:

To rotate the shaft, the linear actuator at the end of the lever arm was activated by a manual directional switch. The linear actuator was extended in two 5 mm stages and then retracted in two 5 mm stages and the break-out torque measured each time. This gave readings Number 1, 2, 3 and 4, as for the linear motion. The whole sequence was then repeated.

Time:

For each force measurement, the time was taken so that the total life of the seals was measured.

7.10.3 Operating Procedure

Before assembly, all of the rig components were thoroughly cleaned with petroleum spirit. The seals were then installed, ensuring that they were greased thoroughly beforehand. The seal rig was then assembled, a suitable thread lubricant such as copper slip being used. Care was taken when the shaft was installed, the threaded ends of the shaft being covered with a plastic cover to prevent them tearing the seals.

The seal rig was pressure tested to 660 bar, just below the rating of the pressure relief valve, in a sealed chamber. No leakage was measured. The rig was then mounted onto the linear actuator support and statically pressurised with water using a hydra pump. Great care was required to ensure that if a seal or fitting failed, the superheated steam which would form when the rig was at 175°C could not cause damage. The actual volume of water used was approximately 0.12 litres, which is very small. A pressure relief valve and an emergency dump valve were mounted on the hydra pump for safety and there was a thermal cut-off to prevent the rig over heating. The basic hydraulic circuit is shown in Figure 7.10.2.

Note that all of the actuation switches were remote from the rig so that the operator did not need to approach the rig while it was at temperature and/or pressure.

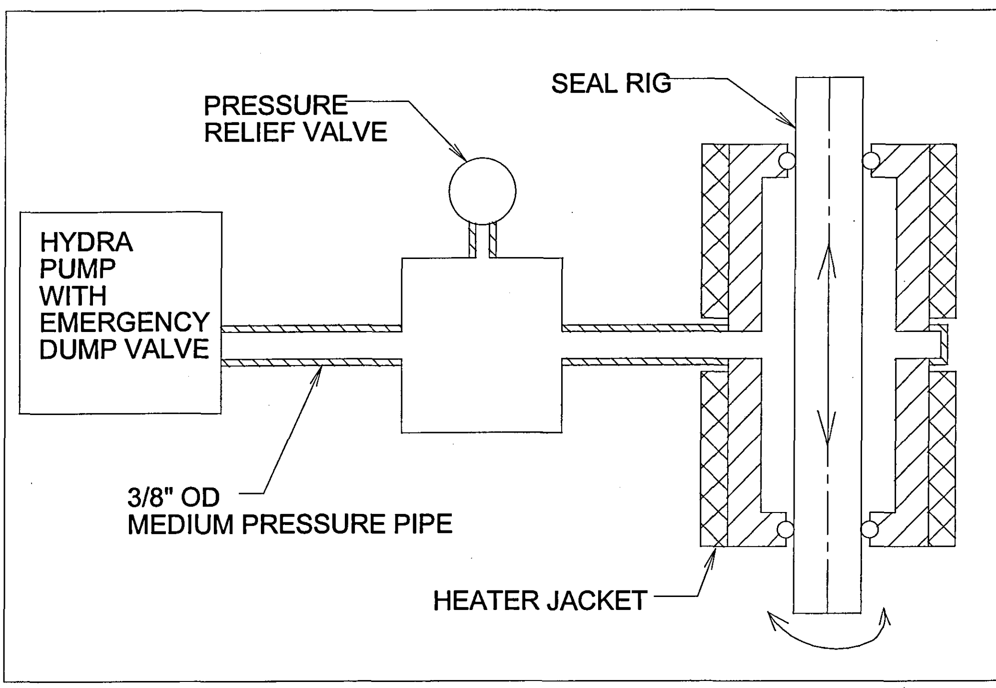


FIGURE 7.10.2 BASIC HYDRAULIC CIRCUIT

The method of operation was:

1) Pressurise to approximately 50 bar to ensure the water stays liquid at high temperature.
2) Heat the system, using heating jackets, to 175°C.
3) Pressurise to the required working pressure of up to 600 bar.
4) Turn off the heaters and measure the linear break-out friction as the test rig cools.
5) Depressurise when cooled to room temperature (<30°C).

The above routine was then repeated to measure the break-out forces in rotation.

7.10.4 Testing

The following four sets of tests were conducted.

Test 1 Linear Actuation At Ambient Temperature and Different Pressures

The ambient rig temperature was taken and the linear break-out force measured for the following pressures:

0, 100, 200, 300, 350, 400, 500 and 600 bar.

Test 2 Rotary Actuation At Ambient Temperature and Different Pressures

The ambient rig temperature was taken and the rotary break-out torque measured for the following pressures:

0, 100, 200, 300, 350, 400, 500 and 600 bar.

Test 3 Linear Actuation At Temperature and Constant Pressure

The linear break-out force was measured at ambient temperature and zero pressure to confirm Test 1. The rig was then pressurised to 50 bar and heated to 175°C. It was then pressurised to 350 bar and the heater switched off to allow the rig to cool. The linear break-out force was measured at 25°C intervals until ambient temperature was reached.

Note that 350 bar was selected because it is the most likely operating pressure.

Test 4 Rotary Actuation At Temperature and Constant Pressure

The rotary break-out torque was measured at ambient temperature and zero pressure to confirm Test 2. The rig was then pressurised to 50 bar and heated to 175°C. It was then pressurised to 350 bar and the heater switched off to allow the rig to cool. The rotary break-out torque was measured at 25°C intervals until ambient temperature was reached.

7.10.5 Results

Before presenting and discussing the results, the purpose of these tests should be reiterated. The aim was to measure the maximum break-out force required to produce linear and rotational motion of a shaft carrying a seal arrangement which is operating at a range of temperatures and pressures. To prove the feasibility of the manipulator design, only significant trends will therefore be discussed.

General

The load cells were approximately sized by the basic predictions given above:

Maximum predicted linear break-out force	16000 N
Maximum predicted rotational break-out torque	11.2 Nm.

The results given in Appendix E (Tests T1, T2, T5 and T6) show that the break-out forces are much lower than these. When the seals were installed and the housing tightened, the frictional forces were too high for the shaft to be moved by hand. After initial movement of the shaft, by using a spanner on one end, the shaft could be rotated by hand.

Note that the measurements for Tests 1 to 4 (a total of nearly 8 hours of testing) were incorrectly taken as only the maximum positive voltmeter readings were measured. Consequently, only the tensile results corresponding to Numbers 3 and 4 are correct. The raw data for these tests is included in Appendix E. All of the tests were repeated and a 'T' in front of the test number indicates that they are correct. Therefore the correct results for Test 1 are given in the table for Test T1.

During the incorrect tests the seals did not leak. Before Test T1 and throughout the other tests, one of the seals leaked at 1 bar. Above this pressure, no leakage was measured and the seals held pressure.

Test T1 Linear Motion at Constant Temperature

The results are plotted on Graph 7.10.1. Note that for an explanation of readings Number 1, 2, 3 and 4 see Section 7.10.2.

There is almost a linear trend of increasing break-out force for increasing pressure. Unsurprisingy, the break-out forces corresponding to Numbers 1 and 3, where the shaft changes its direction of travel, are higher than the break-out forces for Numbers 2 and 4.

A maximum break-out force of approximately 750 N was measured. Note that for the first set of tests, for which Numbers 1 and 2 were incorrect, the maximum break-out force for Number 3 was approximately 950 N at 600 bar. This possibly indicates a reduction in the break-out force as the seals initially become used.

During the linear motion tests, the voltmeter often took several minutes to return to zero. A zero reading was therefore taken before the shaft was moved, although it is not included in the graphs. The time to return to zero indicates the stiction in the seals. The seals take time to return to their initial position after having moved when the shaft was moved.

Test T2 Rotary Motion at Constant Temperature

The results are plotted on Graph 7.10.2.

The break-out torque clearly increases with increasing pressure. Unlike Test T1, there is no indication of a higher break-out torque for Numbers 1 and 3, where the rotation changes direction.

A maximum torque of 1 Nm was measured at 600 bar compared to a maximum of more than 2.5 Nm measured for Number 3 at 600 bar for the first set of tests.

The results are very scattered and there is often a considerable difference between the first and second measurements for each pressure. There are three possible reasons for this:

1) the loads measured are very small compared to the rating of the load cell.

2) the linear actuator force to turn the shaft was minimal compared to the weight of the lever arm (the weight of the lever arm alone was sufficient to turn the shaft).

3) because the weight of the lever arm is so much higher than the loads required to turn the shaft, the load cell reading is sensitive to the position of the lever arm. A slight difference in position at the load cell results in a different zero reading, which corresponds to the load applied to the load cell by the lever arm. Note that a torque of 0.5 Nm is equivalent to a load of 1.1 N at the load cell. The weight of the lever arm is approximately 2 kg.

A combination of these explanations is most likely.

There is therefore a little doubt over the accuracy of the rotary results. However, the maximum torque values, which are the most important, are reasonable. This is because when the rig was pressurised to 600 bar, the shaft could easily be turned by hand without the lever arm. Note that this was not attempted when the rig was at high temperature.

Test T5 Linear Motion at a Pressure of 350 bar and Different Temperatures

The results are plotted on Graph 7.10.3.

Before the seal rig was heated, the break-out friction was measured at zero pressure (this point corresponds to a temperature of 22°C on the Graph). The break-out frictional forces are similar to those measured in Test T1.

As the seal rig cools from 175°C to 150°C, the break-out friction reduces rapidly from a maximum of 900 N to 500 N. The break-out force then gradually drops as the rig cools, until below 100°C the break-out force is almost constant.

Like Test T1, the break-out forces for Numbers 1 and 3 are higher than the forces for Numbers 2 and 4 at the same temperatures.

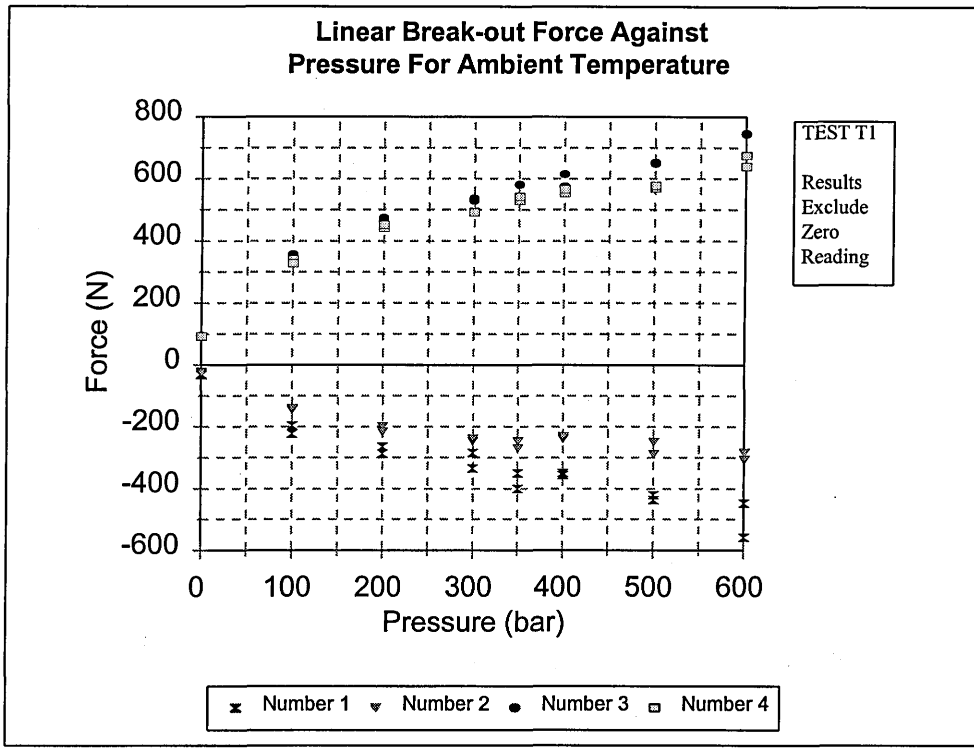
Test T6 Rotary Motion at a Pressure of 350 bar and Different Temperatures

The results are plotted on Graph 7.10.4.

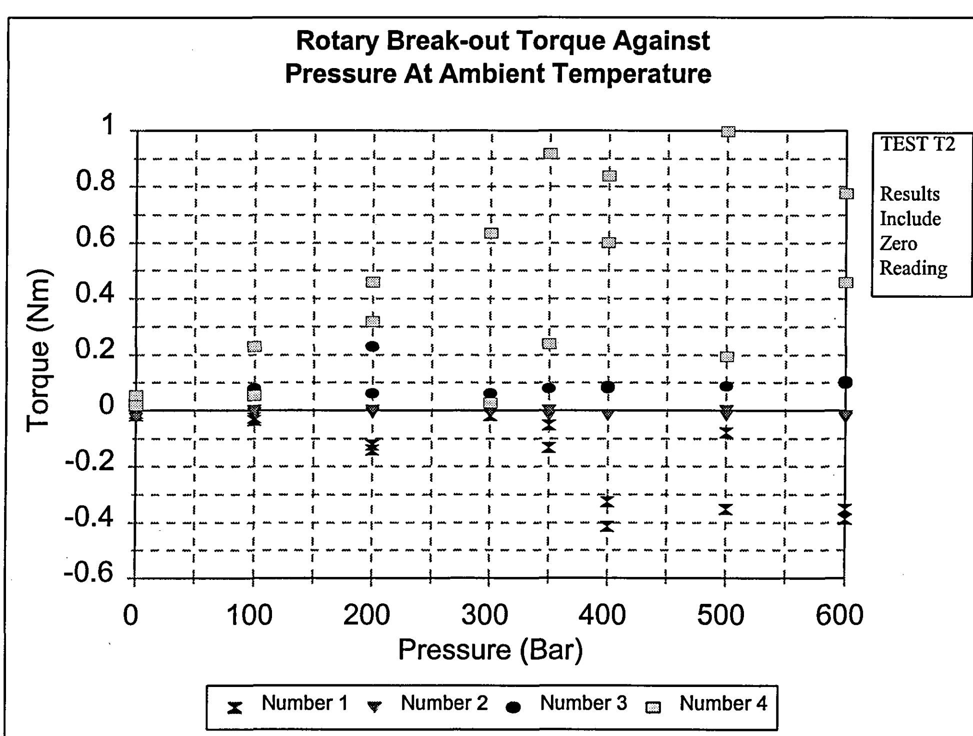
Before the seal rig was heated, the break-out friction was measured at zero pressure (this point corresponds to a temperature of 34°C on the Graph). The measured torques are similar to those measured at zero pressure in Test T2.

There is no clear trend of break-out torque with increasing temperature. The maximum torque is only 0.31 Nm, which is much smaller than the torque measured in Test T2 at ambient temperature and 350 bar.

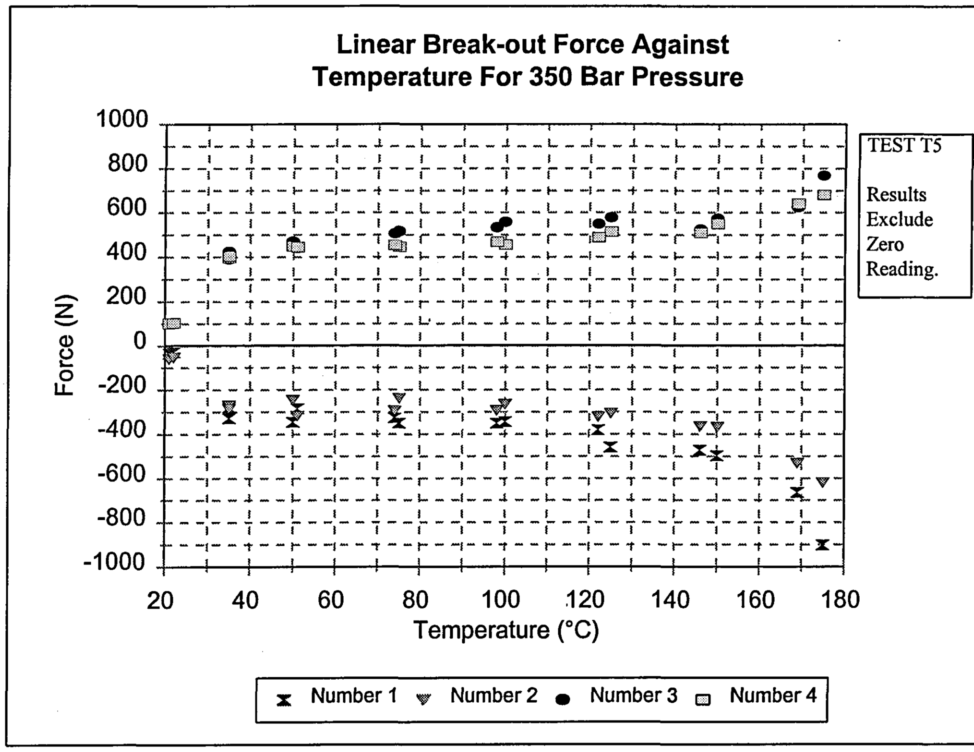
GRAPH 7.10.1



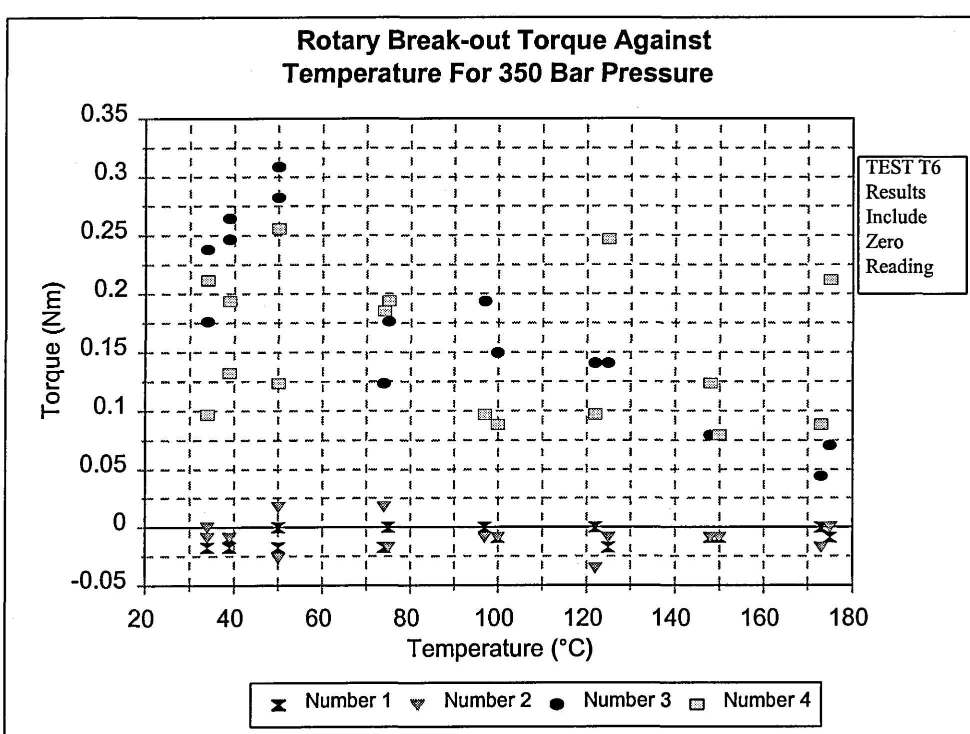
GRAPH 7.10.2



GRAPH 7.10.3



GRAPH 7.10.4



Other Observations

Tests T1, T2, T5 and T6 show that the break-out frictional forces for these seals is very low. Other tests were conducted to obtain additional information. These are summarised below and the raw data is given in Appendix E.

Effect of Changing Between Linear and Rotary Motion

The initial tests, in which the measurements for Numbers 1 and 2 were incorrect, indicated a very high break-out force (as high as 8500 N) when a linear test was conducted after a long period of rotary tests. A detailed set of tests were conducted to investigate this, but no such high results were ever measured. The high break-out force was found to be due to a high voltage reading, incurred while removing the lever arm, which had not been cleared from the voltmeter's memory when the linear force was measured.

Test T3 and T4 Repeat of Tests T1 and T2

After completing the above tests to investigate the effect of changing from rotary to linear motion, Tests T1 and T2 were repeated. For convenience, the rotary tests were conducted without disconnecting the linear actuator from the shaft. The frictional force between the screw threads joining the shaft to the linear actuator was therefore included in the readings. The linear results are the same as T1. The rotary results are similar to T2, indicating that the frictional force due to the screw thread was small (approximately 0.2 Nm), although a maximum torque of 2.5 Nm was measured at 600 bar.

Test T7 Effect of Moving the Shaft for 800 Cycles

At this stage of testing the seals had been operating on plain water for over 23 hours and over 750 movements (375 linear and 375 rotary).

Realistic life trials using abrasive were not possible because:

- there was no method of translating and rotating the shaft automatically.

- there was no method of continually pumping abrasive slurry through the rig.

A crude life test was conducted by using very fine abrasive, which does not settle, and moving the shaft approximately the same number of times as the inner cylinder would have to be moved in the real manipulator (for a 2.5 m long window and a 3 mm diameter nozzle, this corresponds to 830 steps). Note that it is only the fine abrasive which should damage the seals, the wiper should prevent large particles reaching them.

50 μm olivine was added to a small volume of water and the large particles allowed to settle over 5 minutes. The water containing the smallest particles was then poured into the seal test rig. The rig was pressurised to 600 bar and then cycled back and forth, 50 steps linearly and then 50 steps rotationally. The break-out forces were measured regularly during the tests.

Graphs 7.10.6 and 7.10.7 show how the break-out forces changed with increasing number of cycles. The linear break-out force is essentially constant. However, there is a definite increase in break-out torque, to a maximum of 3.5 Nm, as the number of cycles increase. This increase in torque was confirmed by twisting the shaft by hand.

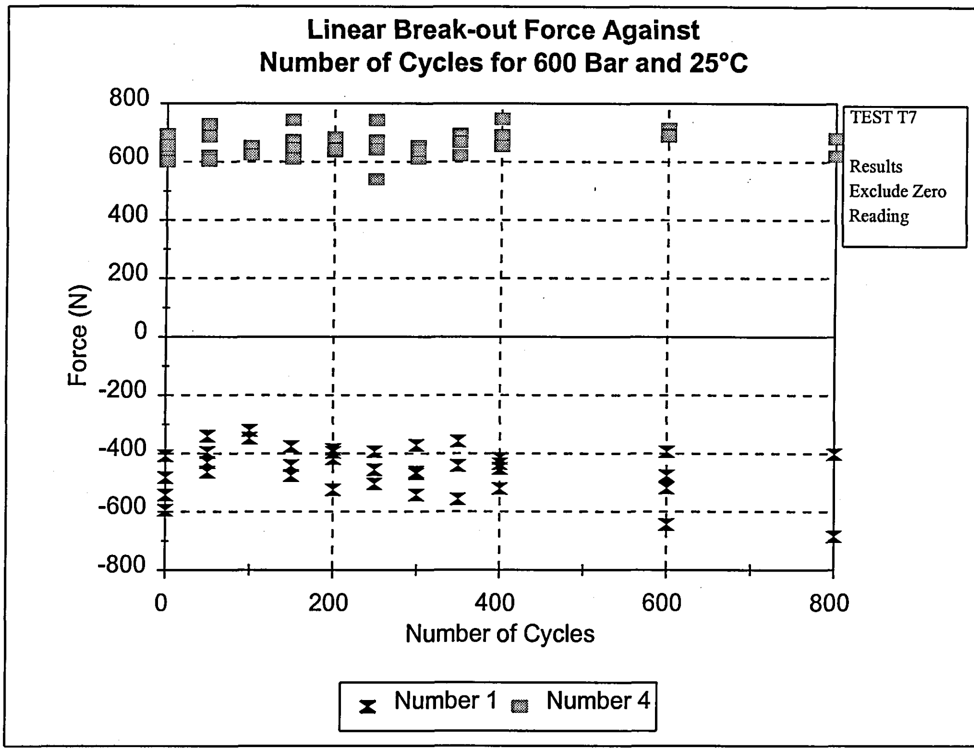
There was no leakage during these tests.

The results indicate that the seals should survive the 20 hours required.

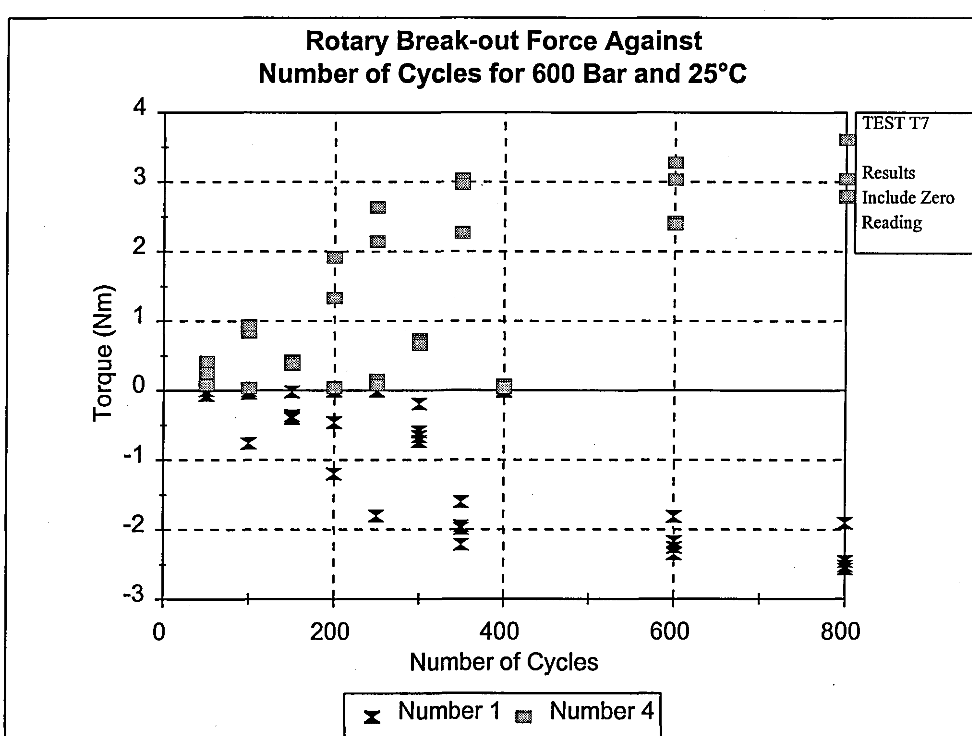
Tests T8 and T9 Repeat T1 and T2 for a New Set of Seals

A new set of wipers and main seals were obtained and Tests T1 and T2 repeated. The results are shown in Graphs 7.10.8 and 7.10.9. The linear results are the same as T1, although a maximum force of 1300 N was measured. The rotary values are much lower than T2 and show no clear trends. All of the results are still much lower than the predicted values.

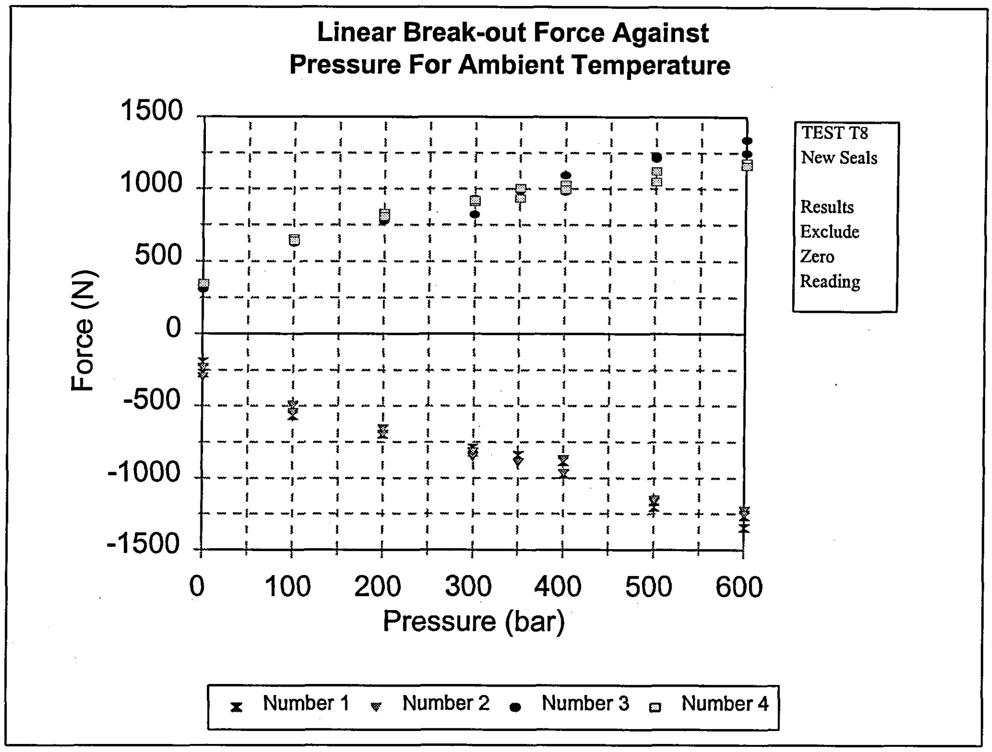
Graph 7.10.6



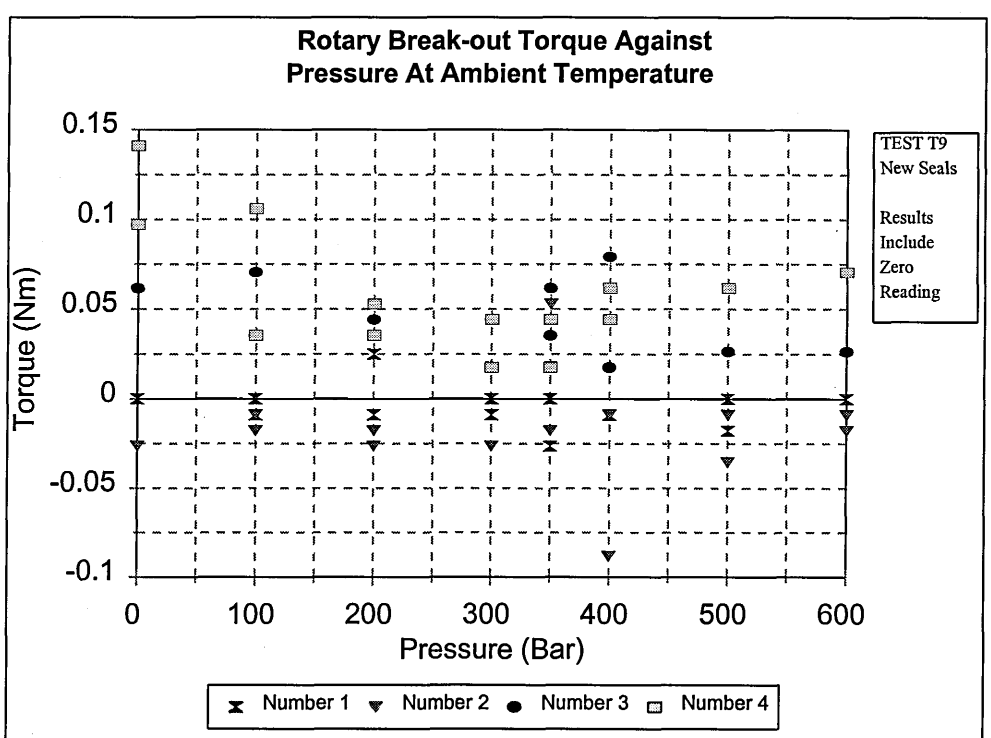
Graph 7.10.7



Graph 7.10.8



Graph 7.10.9

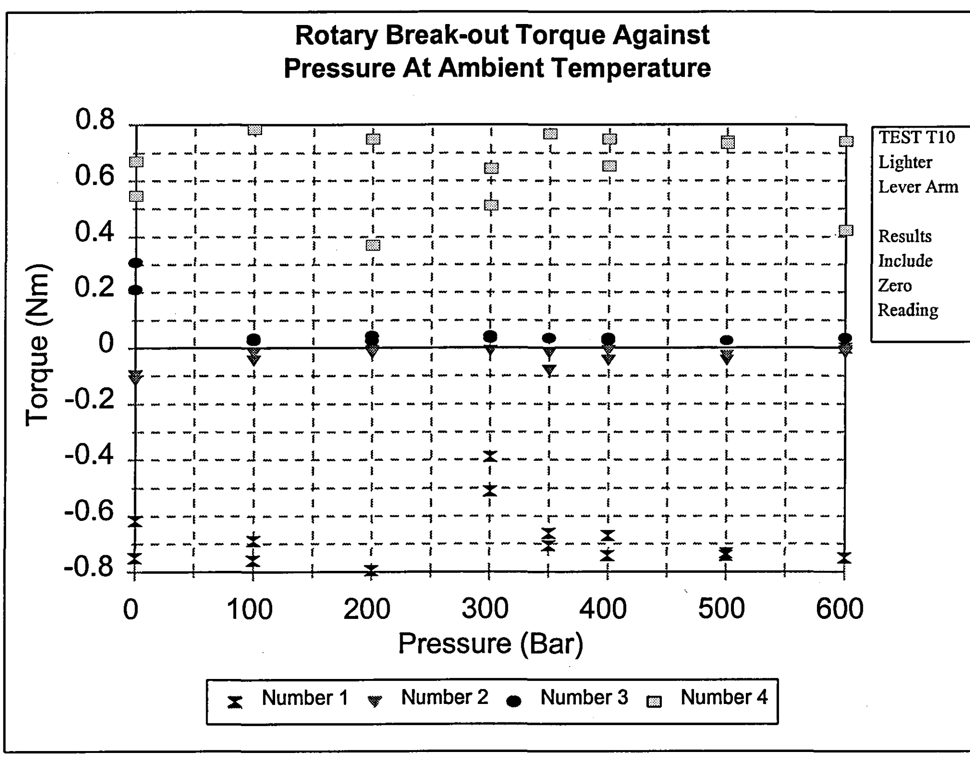


Test T10 Repeat Test T9 With a Lighter Lever Arm

The weight of the lever arm was reduced to 0.6 Kg, by cutting away excess metal using a DIAJET, and Test T9 was repeated. Graph 7.10.10 shows the results, which are similar to the results of T2. The break-out torque for Numbers 1 and 4 are much higher than in Test T9, although the results for Numbers 2 and 3 are very similar. For these results, there is no general trend of increasing break-out torque with increasing pressure.

This test confirms that, although the break-out torque measurements are variable, the maximum break-out torques are consistent and much lower than the predicted torques.

Graph 7.10.10



7.10.6 Conclusion of Seal Tests

Linear and rotational break-out frictional forces were measured for two Zymax composite seals operating at pressures up to 600 bar and temperatures up to 175°C. The forces were considerably lower than predicted such that, once the shaft had been moved after installation, the shaft could be moved by hand. The basic results were confirmed by testing a second set of seals.

For over 20 hours of testing on one set of seals and the various combinations of pressures and temperatures investigated, the results showed that:

> the maximum break-out friction for linear motion = 1300 N
> the maximum break-out friction for rotary motion = 3.5 Nm

A factor of safety of at least two should be used on these results to select the required actuator.

A crude attempt at measuring the life of the seals, when operating in water carrying a suspension of very fine abrasive particles, showed that the break-out torque increased with the number of cycles.

Although one of the seals leaked at 1 bar, there was no leakage during any of the tests.

The seals should therefore be suitable for the manipulator, although they need to be tested in a more realistic abrasive environment.

7.11 The Assembly of the Manipulator

The manipulator must be split in several places to enable each component, including the motors and seals, to be mounted. The exact assembly sequence must be considered very carefully. Figure 7.11.1 shows some of the positions where the manipulator must be split. The exact configuration is considered in the next chapter and Appendix G as part of the detailed design.

The only suitable method of connecting each of these joints is to use some form of screw thread. Screw thread joints of various designs, including metal to metal and elastomeric seals and different locking devices, are used in the oil industry. For the purposes of demonstrating the feasibility of this design, the basic metric thread design and simple 'O' ring will be used. If required, a more suitable thread can be selected during the detailed design.

Accurate machining of the threads is critical. The threads must mate to form almost perfect alignment. Any mis-alignment will build up along the length of the manipulator until the bottom is so mis-aligned that the inner cylinder cannot be moved. Closely toleranced locating shoulders can be machined on one side of the thread so that on engagement, they ensure alignment to within the required tolerance. See Figure 7.11.2. This arrangement was successfully used on the Seal Rig (the distance between the main seals was approximately 190 mm).

Note that, in Figure 7.11.2a, machining the seal and wiper housing into the casing can be difficult, and assembling the seals even more so. The seals can therefore be housed in a seal cartridge, Figure 7.11.2b, which is clamped in position by the outer bodies of the manipulator.

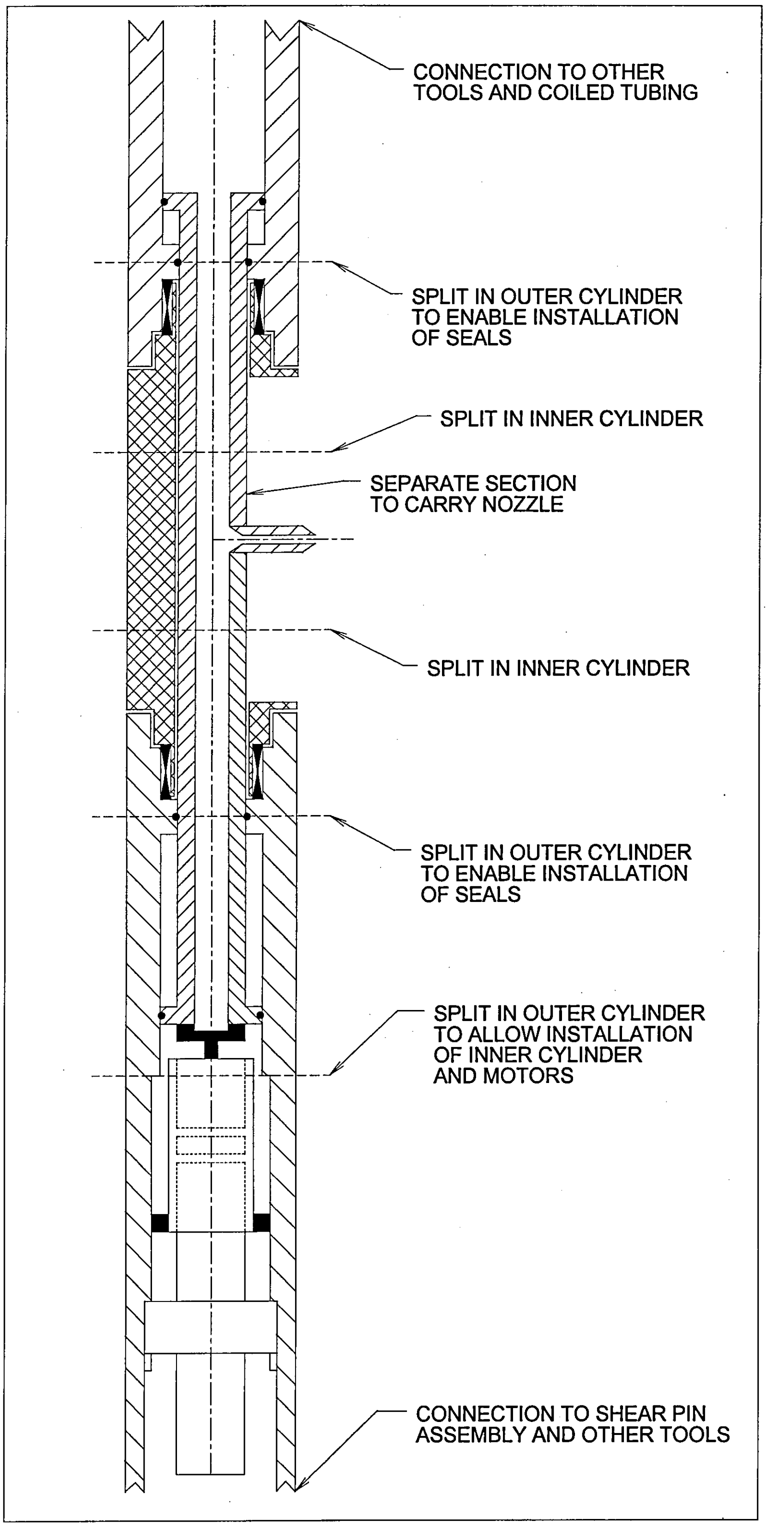


FIGURE 7.11.1 POSSIBLE POSITIONS OF SCREW THREADED CONNECTIONS TO ENABLE ASSEMBLY

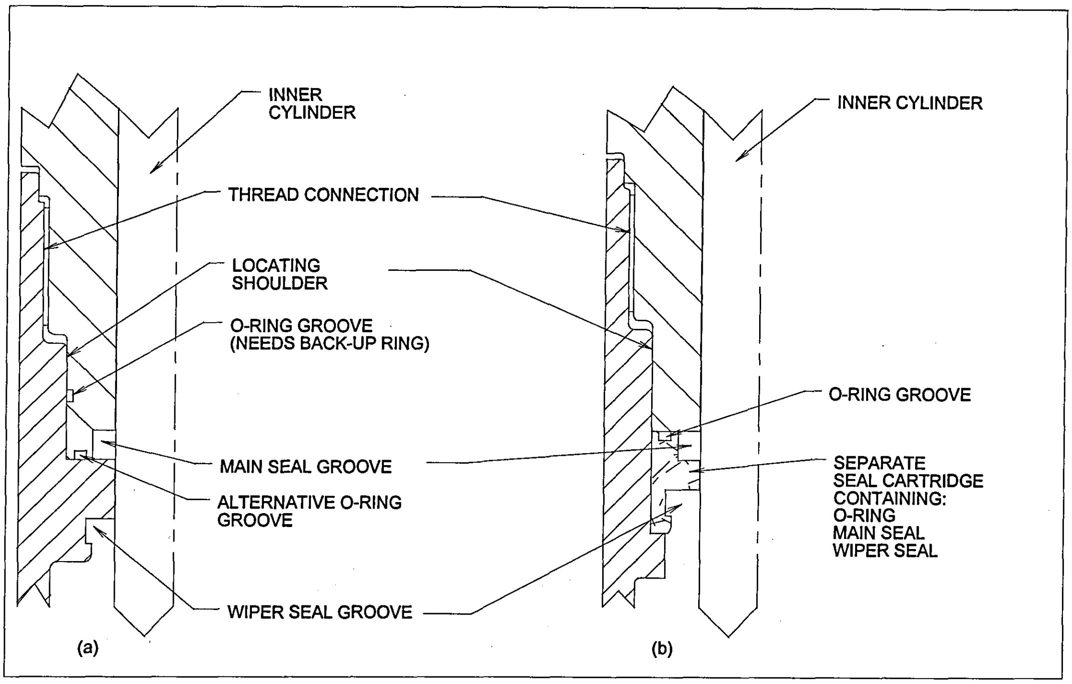


FIGURE 7.11.2 ALTERNATIVE METHODS OF ASSEMBLING AND SEALING THE OUTER CYLINDER, AND INCORPORATING THE MAIN SEALS

7.12 Nozzle Reaction Force

Abrasive water jet cutting is generally considered to be ideal for robotic manipulation because of the low reaction forces. No actual cutting forces need to be applied to the cutting tool. Although this is true for nozzles less than 1.0 mm in diameter, the jet reaction force starts to become significant for larger nozzles.

The general equation for the nozzle reaction force = $C_d.Q.(2\Delta P_N \rho)^{1/2}$

where:	C_d	coefficient of drag through the nozzle = 0.9 for a good nozzle.
	Q	volume flow rate (m^3/s)
	ΔP_N	nozzle pressure drop (N/m^2)
	ρ	fluid density (Kg/m^3).

For $\rho = 1000\ Kg/m^3$, $C_d = 0.9$, Q in l/min, ΔP_N in bar, the nozzle reaction force

$$F = 0.25.Q.(\Delta P_N)^{1/2}$$

Assuming a 3 mm diameter nozzle operating at a nozzle pressure of 350 bar (vol = 100 l/min), then the reaction force would be 470 N. Depending on the length of the stroke, this force could bend the inner cylinder, causing it to lock up.

Some form of support pin will therefore be required which sits between the inner cylinder and the support column, opposite the nozzle. The face of the support pin, which slides along the inside of the support column, must be of a low friction material such as Teflon or PTFE. Abrasive will eventually become embedded in the face of the support pin, causing an increase in the frictional force required to move the inner cylinder. The support pin will therefore need to be regularly examined and replaced if necessary.

7.13 Nozzle Design

The patented DIAJET nozzle consists of a convergent inlet and a straight section of approximately 10 nozzle diameters. If the nozzle is too short the abrasive is accelerated inefficiently and the jet can be incoherent, both of which cause a drop in cutting performance (particularly with stand-off distance). If the nozzle is too long, the frictional forces in the nozzle become too high and the cutting fluid and abrasive slow down, again reducing the cutting performance.

The maximum outside diameter of the manipulator is 53 mm. Considering that nozzles up to 3 mm in diameter are going to be used, the nozzle must be over 30 mm in length.

If such a nozzle was incorporated into the inner cylinder, the nozzle would partially block the bore of the manipulator resulting in both an increased pressure loss and increased wear. A shorter nozzle must therefore be used (see Appendix G3).

Cutting trials were conducted to investigate the effect nozzle length has on cutting performance. Two 0.5 mm diameter tungsten carbide nozzles were made with diameter to length ratios of 6 and 10, and the maximum speed to cut 12 mm thick stainless steel was determined for each nozzle. The tests found that a nozzle with a diameter to length ratio of 6 cut at approximately 90% of the parting speed for a standard nozzle, with a diameter to length ratio of 10. This slight reduction should be more than covered by the safety factors in the window cutting prediction program (see Section 5.4).

Throughout the window cutting operation, it is important that the operators know exactly where the nozzle is. This should be possible from the encoder feedback from the motors. However, if the nozzle is not correctly orientated in the well it must first be pointed towards the casing wall. Standard tools, such as gyroscopes, are used regularly in the oil industry to ensure the correct orientation of downhole tools.

7.14 The Flow Velocity Through The Inner Cylinder

The inside diameter of the inner cylinder must be as large as possible. However, from simple stress calculations the outside diameter of the inner cylinder will be approximately 20 mm and the inner diameter approximately 10 - 12 mm. Assuming 100 l/min of abrasive fluid has to pass down through the inner cylinder, the flow velocity will be in the order of 20 m/s. The bore of the inner cylinder must therefore be protected by using a thin walled liner which carries no pressure loading.

7.14.1 Simple Wear Tests

Simple wear tests were conducted on 316 stainless steel test pieces, as shown in Figure 7.14.1. A low pressure slurry was continuously pumped through the wear piece using a slurry pump and the diameter of the bore was measured regularly. The abrasive flow rate varied between 0.6 and 2.3 Kg/min (in 2.9 to 3.3 l/min of water respectively) so that abrasive velocities were between 15 and 22 m/s. Note that the abrasive concentration is over twice the concentration which would be used for cutting downhole.

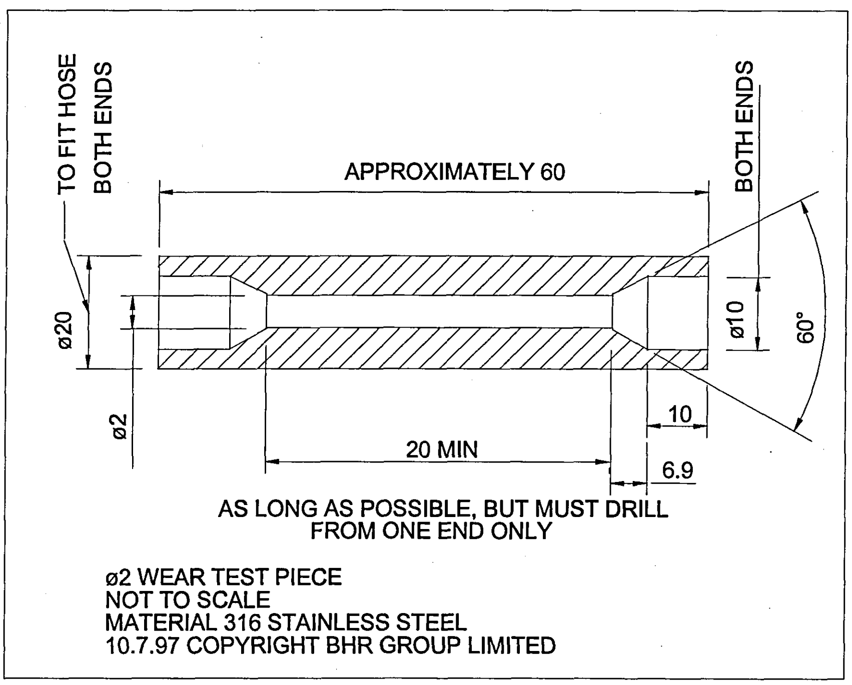


FIGURE 7.14.1 STAINLESS STEEL WEAR PIECE

The wear piece wore rapidly such that after 9 hours of testing the maximum diameter was 3.5 mm, a reduction in wall thickness of 0.75 mm. Note that the pump flow was increased as the bore grew to maintain the high velocities.

This test was repeated with a 316 test piece which had first been surface hardened to 60 RC by Kolsterizing (see Section 7.10.1). Unfortunately no reduction in wear rate was measured, the 33 μm hardened layer being removed too quickly for any difference to be measured. Other test pieces run with an abrasive slurry velocity of only 6 m/s showed that the Kolsterized wear piece took over 5 times as long to wear.

This shows that a thin walled liner can be used to protect the inner cylinder, but it must be made of a material which can be through hardened to over 60 RC if it is to last the required 20 hours in abrasive velocities of over 20 m/s.

As an electrical cable has to be installed inside the cylinder, the velocity will increase further. The electrical cable must similarly be protected, by either installing it in a separate long metal sleeve (provided the sleeve passes below the nozzle there should be no flow through it) or by hard surfacing the cable in some way. The diameter of the cable is critical and must be as small as possible. Some form of junction box will therefore be needed above the inner cylinder. The large diameter wireline cable will connect to the junction box and much smaller diameter cables will feed from it through the inner cylinder to the motors.

7.15 Flushing the Coiled Tubing

The Cutting Head experiments detailed in Section 7.8 showed that if the cutting fluid does not remove all of the abrasive from around the manipulator, a method of flushing it from the well was needed. To achieve this, a second port is required. Abrasive free cutting fluid can then be pumped through both the nozzle and the second port to flush the well clean.

This second port must be sealed during cutting. A valve is therefore needed which can fit into the space available. It must be suitable for operating in an abrasive environment and seal at the operating pressures. Ideally the port diameter should be at least as big as the nozzle diameter. The most suitable type, of valve for this application is a plug type valve, as shown in Figure 7.15.1 (Ref 106). This could be solenoid actuated.

The valve should, theoretically, be exposed to minimal quantities of abrasive. When the well needs to be cleaned, cutting is stopped and abrasive is flushed out of the coiled tubing through the nozzle. Once no abrasive is left in the coiled tubing, the valve can be opened and abrasive free fluid pumped through the nozzle and second port. Once the well has been sufficiently cleaned, the valve is closed, again operating on abrasive free fluid only, and cutting can be resumed.

The challenge is to provide a method of actuating the valve, which depends on whether there is a pressure difference across it. Consider a plug valve which seals on a 4 mm diameter seat and which needs to be opened at a pressure differential of 350 bar. The actuation force required to do this is approximately 440 N. Alternatively, the required actuation force will be minimal if there is sufficient time to allow the coiled tubing to be depressurised first.

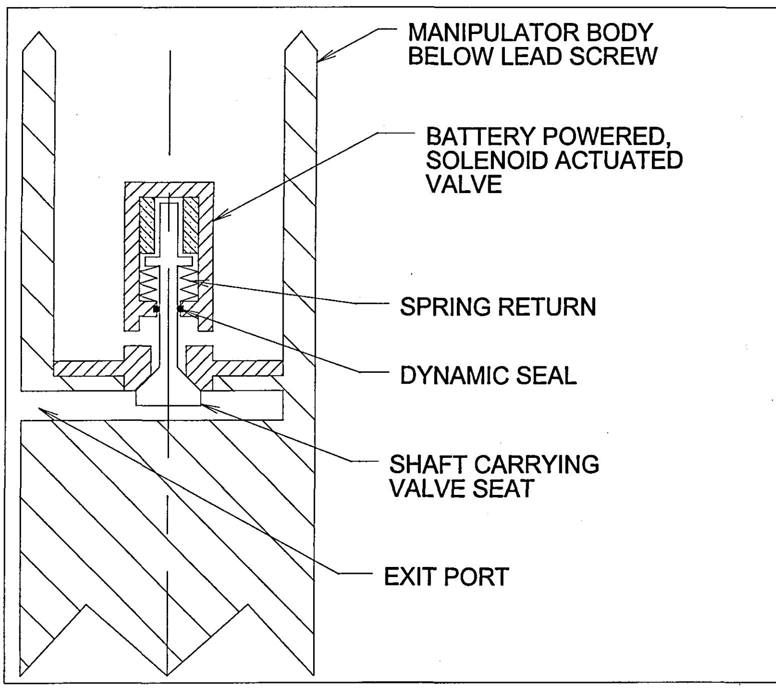


FIGURE 7.15.1 A SCHEMATIC FLUSHING VALVE

The only feasible position for this valve is below the lead screw. Unfortunately, passing an electrical cable through or around the lead screw nut to power the solenoid is difficult because of the very limited space available. There are two possible alternatives to actuating the valve:

1) If possible, the solenoid could be powered by a battery, although an actuation force as high as 440 N may not be possible. A method of operating the valve is then needed. This could be crudely achieved by moving the end of the lead screw down until it meets a switch to operate the valve. Note that the switch would have to be pressure balanced so that it would not actuate at pressure. To close the valve, the lead screw is moved up and the valve shuts either by a spring return or by reversing the polarity of the solenoid. Once closed, the valve will stay closed once the manipulator is pressurised.

2) Alternatively, the lead screw could be the actuator. The lead screw could be lowered onto the top of the valve, pushing the valve open. When the lead screw is raised, the valve will close by spring return.

Note that this valve is **not** for emergencies. If the lead screw motor failed, the valve would not open. However, there is no real need for an emergency valve in the manipulator. If the coiled tubing becomes over pressurised, perhaps if the nozzle has blocked, a pressure relief valve, located at the surface, would open and vent the pressure.

Provided the cutting fluid can suspend the abrasive adequately and the system has been fully depressurised, the manipulator can be retrieved without first having to flush the coiled tubing clear of abrasive. The weight of the abrasive fluid in the coiled tubing is balanced, due to buoyancy, by the abrasive fluid in the well. Consequently, only a minor increase in pull load is required to lift the coiled tubing out of the well.

7.16 Summary

This chapter presented the basic manipulator development and identified the critical components associated with it. In particular, a model cutting head and a seal test rig were built, which confirmed that the basic design is feasible. A final sketch of the main manipulator components is shown in Figure 7.16.1.

Some areas of concern have not been answered satisfactorily. These are discussed in Chapter 9, Future Work, but will really only be fully addressed when a working manipulator is built.

The next chapter discusses the detailed design of the manipulator. To keep the chapter concise, components which are conventional to the oil industry have not been designed.

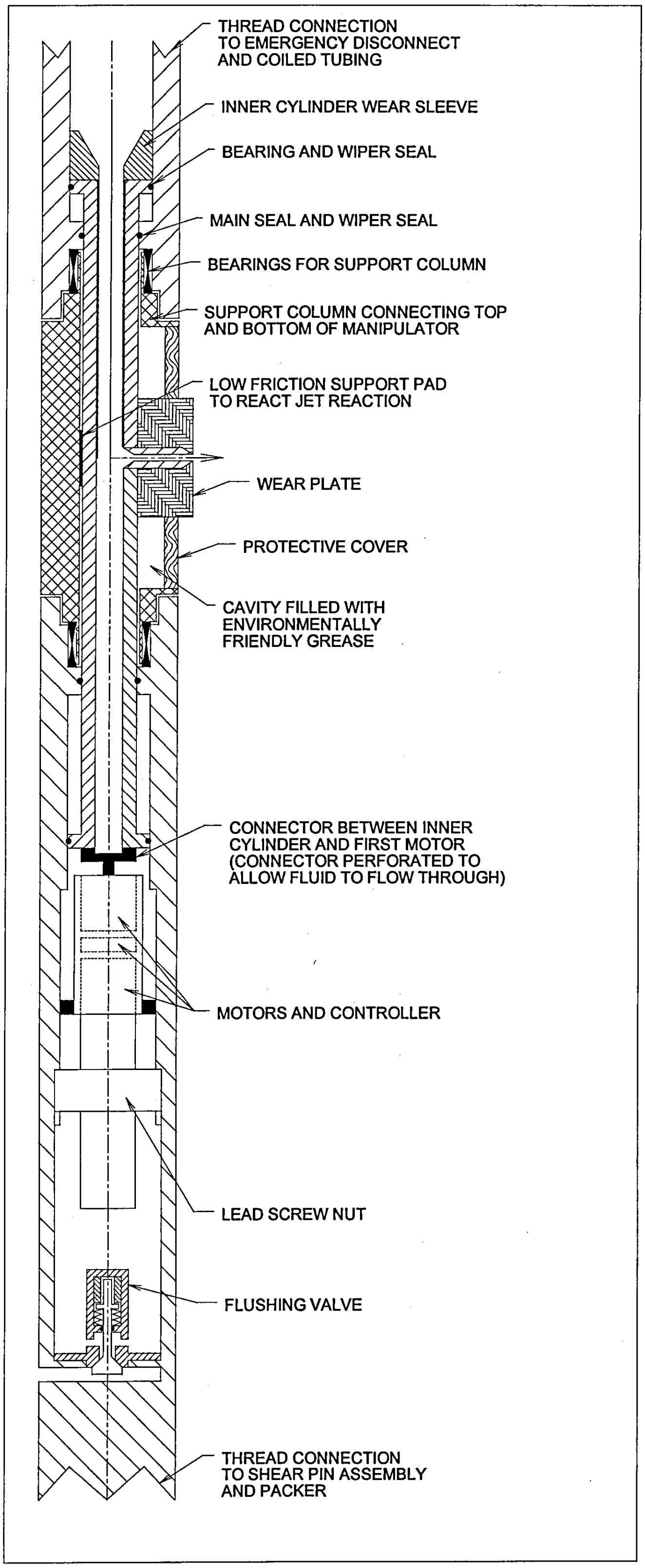


FIGURE 7.16.1 THE MAIN ASSEMBLY OF AN ABRASIVE WATER JET WINDOW CUTTING MANIPULATOR

8.0 DETAILED DESIGN

8.1 Introduction

This chapter presents the final manipulator design, based on simple stress calculations and standard available components, including material selection, and details the method of assembling the manipulator.

To reach this final stage, many iterations of the basic calculations were required, each iteration usually failing due to the size restrictions or manufacturing considerations. Consequently the basic design development shown in Chapter 7 progressed simultaneously with the basic stressing presented here. The two processes have been separated in this thesis to more clearly present the design considerations. Only the final design calculations are given, and these are presented in Appendix G.

For the purposes of this feasibility study, detailed stressing is not required. Simple stress calculations and large factors of safety are sufficient. The basic manipulator design presented here is likely to have several versions depending on the exact cutting requirement and well conditions. It is only when these conditions are known that detailed stress calculations need to be undertaken.

8.2 Overview of the Basic Stress Calculations

8.2.1 Pressure Vessel Design

The manipulator body must be correctly sized to withstand the high internal pressures. Standard pressure vessel design codes, such as BS5500 (Ref 107), are generally applicable. However, for the very high pressures we are using, the pressure vessel wall thickness needs to be relatively large. BS5500 only applies to thin walled vessels where:

$$\frac{OD}{ID} < 1.3$$

where: OD - outer diameter
ID - inner diameter

An alternative relationship is required for thick walled pressure vessels. This is derived in Appendix G, and is based on applying equations for the principal stresses for a hollow cylinder under uniform internal pressure with the ends capped (Ref 108) to Von Mises equation for failure (theory of constant energy of distortion).

The final relationship is:

$$\sigma_{eff} = \sqrt{3}\ \frac{(OD)^2}{((OD)^2 - (ID)^2)}\quad P \qquad 8.1$$

σ_{eff} - effective stress N/m^2
P - maximum internal design pressure N/m^2

For no failure, the material design stress f must be greater than $\sigma_{eff.}$ The material design stress is discussed below.

8.2.2 Material Design Stress

BS 5500 (Ref 107, Section K/2) describes in detail different design stresses, depending on the type of material. For example, for Austenitic stainless steels above 150°C,

$$f = \frac{Re\ (T)}{1.35} \text{ or } \frac{Rm}{2.5}$$

whichever gives the lower value.

Where Re(T) is the minimum specified yield strength at the operating temperature T,
Rm is the minimum tensile strength at room temperature.

The final material used will depend on many factors apart from the design stress. Material selection is discussed in detail later in the chapter. Instead of using the BS 5500 definition for design stress a simpler alternative is to define f as 60% of the minimum specified yield strength at the operating temperature T.

$$f = \frac{Re(T)}{1.67}$$

This gives a lower value of f than BS 5500 and can be applied to the range of materials that could be considered.

Table 8.2.1 gives a range of f values for an operating temperature of 200°C (note that the maximum operating temperature defined in the Specification, Chapter 6, was 175°C).

Initial stress calculations soon showed that design stresses in the order of 250 MN/m^2 were required. **All of the following design calculations are therefore based on an initial design stress of 250 MN/m^2.**

TABLE 8.2.1 DESIGN STRESSES FOR DIFFERENT STEELS

Type of Steel	Yield Stress At Room Temperature MN/m^2	Yield Stress at 200°C MN/m^2	f at 200°C MN/m^2	Reference
316 S-11 stainless steel	225	165	100	BS 5500 (Ref 109)
Ferralium alloy (255 SD40) (Duplex stainless steel)	550	400	240	Meighs Alloys (Ref 110)
17-4 PH (S17400 630 H1150)	885	575 (approx.)	345 (approx.)	Ugine Savoie (Ref 111)
EN24T Chromium-Molybdenum Steel	800	770	462	Macreadys Carbon + Alloy Steels (Ref 112)

8.2.3 Thread Stressing

Section 7.9 explains that the manipulator must consist of several sections, coupled together by screw threads. The Seal Rig showed that this was feasible, although some form of locking device, such as a grub screw, is required on the manipulator threads to ensure that they do not loosen.

Conducting thread stress calculations, especially fatigue calculations, is very difficult because the care and precision of their manufacture is a critical factor; each thread root is a stress concentration and a potential site for a crack. As previously mentioned, there are a range of different thread designs used in the oil industry which have been optimised to give the best strength and fatigue characteristics. An expert would be able to select the most appropriate one for this application.

Only very basic stress calculations have therefore been conducted to select the appropriate size of metric thread. These are based on the equations presented in Reference 113, which check that there is sufficient material to carry the direct stresses and that the threads are not "stripped". Note that the equations have been modified to include the material design stress.

The component tensile force, TF, required to yield the entire threaded cross section is:

$$TF = A_x\, f \qquad 8.2$$

where A is the critical direct stress area of component x.

For female components:

$$A_f = \frac{\pi}{4} [\text{diameter internal thread}]^2$$

For male components:

$$A_m = \frac{\pi}{4} [\text{minor diameter external thread}]^2$$

The Industrial Fasteners Handbook (Ref 114) lists the basic dimensions for the coarse series and fine series of ISO Metric Threads. Alternatively, the minor diameter for the internal thread (MDIT) can be calculated from the expression:

$$\text{MDIT} = \text{Basic thread diameter} - 2\left(\frac{5}{8} * 0.86603p\right) \qquad 8.3$$

where: p is the thread pitch.

The tensile force required to yield the entire thread-stripping failure surface is:

$$TF = \pi d\,(0.75t)\, f_s$$

where: $\pi d\,(0.75t)$ is the shear area,
d is the thread diameter (the diameter of the shear fracture surface)
t is the minimum length of thread
f_s is the design shear stress

Reference 113 shows that the yield stress in shear (shear yield strength) for ductile materials is approximately 0.58 times the direct yield stress. Consequently a design stress in shear, f_s, will be defined as:

$$f_s = 0.58f$$

$$\therefore \quad TF = \pi d\,(0.75t)\,(0.58f)$$

$$TF = 0.435\, \pi d f \qquad 8.4$$

By equating equations 8.2 and 8.4, the length of thread can be calculated for which tensile failure and thread stripping will occur at the same time.

For stressing nuts and bolts, the nut is generally made of a softer material to allow slight yielding of the top threads, allowing the load to be distributed more evenly. Standard nut thicknesses are approximately ⅞d (Ref 113).

The thread roots act as stress concentrations. A stress concentration factor should therefore be included in the applied load. Ref 113 specifies a stress concentration factor, k_t, of 2.8 for machined threads. The thread should therefore be designed to carry 2.8 times the expected applied load. This has important implications on the final design. In some cases the thickness of material required to satisfy this condition means that there is no space for the cutting fluid. Instead of applying k_t, the maximum load capacity of the thread will be calculated and compared with the working load. This ratio can then be compared with k_t.

The manipulator will only be used in 20 hour spells. Critical threads can be ultrasonically inspected after each operation. Such a high value of k_t should not, therefore, be necessary.

For the thread calculations presented, the minimum length of thread is calculated using equation 8.4. If required, this can be increased to ⅞d, as the length of the manipulator is not restricted.

8.3 Applied Loads

Before the manipulator can be stressed, the applied loads need to be identified. The main loads are due to:

- the pressure forces.
- the forces associated with introducing, and removing, the manipulator into the well. Note that during these procedures, there is no internal pressure.
- frictional forces associated with moving the internal cylinder.
- the forces associated with maintaining the position of the manipulator - these react the pressure forces and the frictional forces.

8.3.1 Working Pressure and Design Pressure

Section 5.4 showed that 1½" coiled tubing is likely to be used for this application. This has a surface pressure limit of 595 bar, assuming a 20% safety factor. There is therefore a possibility that the manipulator could be operated at 595 bar. In general, because of the pressure losses, 350 bar will be more typical.

There is some confusion over a suitable safety factor to calculate the design pressure. BS5500 (Ref 107) does not specify a safety factor, the design stress already accounting for it. However, BS 5500 does state that the pressure vessel should be designed to the pressure rating of the pressure relief valve. This is usually between 10% and 20% over the working pressure. Other rules of thumb used in the jetting industry quote factors from 1.25 to 1.5 times the working pressure.

A factor as high as 1.5 would have serious implications on the size of the manipulator. A factor of 1.25 will therefore be used.

Maximum Working pressure (MWP) = 595 bar

Maximum Design pressure (MDP) = 744 bar

Note that if the expected working pressure is 350 bar, this gives a factor of safety of over 2 for the manipulator stress calculations. This is important in terms of the component fatigue lives.

8.3.2 Reaction Forces

The coiled tubing must be kept in tension to maintain the position of the nozzle. This force, RF, is:

$$RF = ID \times MDP$$

where ID is the internal diameter of the manipulator body.

8.3.3 Removing the manipulator from the well

The maximum force applied to the manipulator when it is removed from the well is the pull capability of the coiled tubing. For 1½" coiled tubing, 0.156" wall, the minimum yield load capacity is 205,053 N (Ref 115). This is very large. The manipulator will therefore be designed based on the operating conditions and the final strength then compared with this load capacity. It is likely that the manipulator will fail before the coiled tubing load capacity is reached.

8.3.4 Frictional Forces

Section 7.10 predicts that the maximum translational forces due to seal friction will be 16,000 N, although only 1300 N was measured on the seal rig. For design purposes, 15,000 N will be assumed. 11.2 Nm will be assumed to be the maximum applied torque (see Section 7.9).

8.4 Manipulator Body Design

The main part of the manipulator, shown in Figure 7.16.1, consists of :

- an outer cylinder (upper section and lower section) which carries the main seals.
- two seal cartridges.
- a support column, with a bearing arrangement at either end, to hold the manipulator together and maintain the alignment of the upper and lower sections of the outer cylinder.
- a moveable inner cylinder which is situated within the outer cylinder and support column and is protected by a liner.
- a nozzle holder, which is connected to the inner cylinder. The nozzle holder carries the nozzle and wear plate assembly.
- a motor and lead screw assembly which connect to the lower part of the inner cylinder. The lead screw passes through a nut connected to the outer cylinder.
- a flushing value.
- a shear pin assembly.

The simple design flow chart below (Table 8.4) shows the basic design stages which are detailed in Appendix G. If required, Appendix G should now be studied.

TABLE 8.4 MANIPULATOR DESIGN FLOW CHART FOR APPENDIX G

COMPONENT	OPERATION	RESULT
Outer Cylinder Design.	Based on the maximum OD of the manipulator and seal dimensions quoted by James Walker, size the outer cylinder to carry the pressure.	Maximum ID of the outer cylinder and therefore the maximum OD of the inner cylinder.
	Select appropriate thread and seal arrangement for the outer cylinder connection.	
Inner Cylinder Design.	Select the appropriate connection for the inner cylinder to the nozzle holder, including grub screws to react the torque. The method of assembly must be considered.	Maximum ID of the inner cylinder and the minimum OD of the nozzle holder.
	Inner cylinder wear protection.	
	Buckling of inner cylinder.	Maximum length of inner cylinder.
Nozzle Holder Design.	Design nozzle assembly and include space for the wear plate. Note: the nozzle assembly must not protrude beyond the maximum allowable diameter.	Length of nozzle and wear plate.
Support Column Design.	Consider the loads applied to the support column and confirm the basic design is acceptable. This is based on the inner cylinder and nozzle holder design.	Confirm column dimensions.
	Bearing selection.	
	Bearing assembly design.	
	Loading pin design to apply tensile loads to the support column.	
	Calculation of bearing friction.	Minimum torque required to turn the support column and inner cylinder.
Lead Screw Selection.	The stroke of the inner cylinder is determined by the dimensions of the support column, nozzle holder and maximum length of the inner cylinder.	Length of the lead screw.
	Select the lead screw diameter based on the buckling load.	Final lead screw dimensions.
	Calculate the torque required to turn the lead screw.	Minimum torque required to turn the lead screw, support column and inner cylinder.
Motor Selection.	Based on the final torque requirements, select motors and gearbox.	Final motor selection.
	Design motor housing and anti-rotation pins.	
Lower Outer Cylinder Design.	Design the housing to incorporate the lead screw nut, including ports to allow flow around the nut.	
	Consider the incorporation of the flushing valve.	

8.5 Materials Selection

The manipulator has been stressed assuming a minimum material design stress of 250 MN/m^2.

The stresses calculated in Appendix G are below this design stress, except for the outer cylinder which needs a minimum design stress of 262 MN/m^2. However, the safety factors on the screw threads are generally lower than the required 2.8, although there should not be a problem under normal working conditions. If the manipulator is operated for long periods above 350 bar, greater care should be taken, and the threads should be examined regularly.

There are other material properties which are important for this application, and these need to be identified. Material selection will therefore be approached in the following two ways:

1) A general materials selection check list will be covered to prioritise the most important material characteristics.

2) For each of the main manipulator components, the critical material requirements will be identified.

From this, a range of potential materials can be identified and the most appropriate selected.

8.5.1 General Materials Selection Checklist

Mechanical

Material Property	Category of Importance	Comments
Design Strength	1	To satisfy basic design, f > 250 MN/m^2 at operating temperature 200°C.
Modulus	2	Small changes in shape are not serious.
Impact Strength	1	Will experience impacts and jarring during introduction to and withdrawal from the well.
Fatigue Strength	1 / 2	Component will be used for maximum of 20 hours before maintenance. NDT checks can be conducted on critical components every 20 hours. The manipulator is more likely to be damaged in handling rather than fatigue. Fatigue failure of threads is critical, especially for the outer cylinder. Failure would make removing the manipulator from the well very difficult.
Creep Strength	5	Operating temperature below critical temperature for creep, which will be approximately 500°C.
Hardness	1 / 2	On sealing surfaces, hardness and surface roughness are very important. A minimum hardness of 40 RC is specified by seal suppliers. Some components are exposed to abrasive wear. A surface hardness of over 60 RC has been found to significantly increase component life.
Fracture Toughness	1	Fast fracture at pressure, particularly during pressure testing, is undesirable.
Machinability	1	Machining costs will be >> than material costs.

Environmental

Conditions	Category of Importance	Comments
Atmospheric Corrosion	1 / 2	Possible short term exposure to water (rain) and sea water. With careful storage and cleaning this may not be a problem. Possibility of pitting of sealing surface.
Aqueous Corrosion	1 / 2	Component exposed to well fluids although only for a short time. Chemical content e.g. H_2S, O_2, NaCl etc will determine corrosion severity. Possibility of pitting of sealing surface. In general, material requirements for good corrosion properties will depend upon chemical content. Different materials would be used for different environmental conditions.
Cutting Fluid	1	The cutting fluid will contain polymer additives to enhance solids carrying and reduce pressure losses. Corrosion inhibitors could be added to protect the bore of the manipulator (particularly the bore of the outer cylinder).
High Temperature	1	All material properties must satisfy minimum design stresses at 200°C.
Stress Corrosion Cracking	1	Corrosive chemicals and stress present.

Key to Category of Importance:

1 - very important. 5 - of least importance.

Economic

	Category of Importance	Comments
Material Costs	2	
Fabrication Costs	2	Ease of assembly very important.
"Cost of Ownership"	2	Must be competitive with alternate methods, but these costs must be considered as part of the total operating cost.
Recycling	2	If the manipulator is handled with care, with critical sections examined regularly, the manipulator can be reused many times. Only the consumable components, such as the nozzle, wear plate and bearings, need to be replaced. Recycling of the manipulator at the end of its working life is less important.

Aesthetics

	Category of Importance	Comments
Surface Finish	1 / 2	Depends on component.
Colour	3	Final manipulator needs to "look good"

Physical

	Category of Importance	Comments
Conduction	5	Not important
Expansion	5	Not important
Phase Change	5	Not important
Electrical Properties	5	Not important
Magnetic Properties	3	May be important if manipulator is used with certain sensors.
Density	5	Not important

Safety

	Category of Importance	Comments
Toxicity	5	Not important
Flammability	1	Could be present in flammable environment.

Key to Category of Importance:

1 - very important. 5 - of least importance.

8.5.2 Manipulator Components and Their Critical Design Features

Component	Design Feature	Comments
Outer Cylinder	Screw threads: Machinability No galling Corrosion resistance Inner bore: surface finish for minimum friction with bearing and wiper seals (Ra < 4 μm).	f > 262 MN/m^2 for pressure retension. Required stress concentration factor on threads = 2.8. Current factor = 1.46.
Inner Cylinder	Screw threads Outer surface: need good surface finish (Ra < 4 μm) Sealing surface must not corrode.	f > 250 MN/m^2. Factor on threads = 1.4. Need surface hardness > 40 RC.
Support Column	Wall thickness at bearings	f > 250 MN/m^2. Factor on threads = 2.8.
Loading Pins	Shear failure	Pin shear stress at 200°C, f_s > 393 MN/m^2.
Inner Cylinder Wear Liner	Abrasive erosion of inner bore; abrasive velocity could be as high as 25 m/s.	Liner must last over 20 hours without failure. Inner bore should be hardened to over 60 RC.
Nozzle Holder	Screw threads	f > 250 MN/m^2. Factor on threads = 2.36.
Motor Housing	Thin wall must react applied torques.	f > 250 MN/m^2.
Wear Plate	Abrasive erosion from the direct impact of the reflected jet.	Wear plate must last over 20 hours without failing. Surface hardness should be > 60 RC.

8.5.3 Possible Materials

High strength steels are the only possible materials for most of the manipulator components. The most suitable are :

Chromium Molybdenum Steel and High Strength Carbon Steels

A chromium molybdenum steel, such as EN24T has a very high design stress (462 MN/m^2) and is ideal for screw threads and pressure retension. Corrosion resistance is very poor and protection would be required. However, exposure to a corrosive environment is for only short periods. Provided the components are carefully cleaned and protected after use, there should be no problem for most applications.

For example, EN24T was used for the main body of the seal rig. The seal tests, conducted at up to 600 bar and 175°C, were performed over a period of eight weeks. Water was left in the rig during the whole time and there was no sign of corrosion.

Although these steels can be hardened, the 40RC required is too high to achieve. They would not be suitable for the inner cylinder.

These steels would be particularly suitable for the outer cylinder, seal cartridge, support column and nozzle holder. All of the outer surfaces should be protected.

Ferralium (Duplex Stainless Steel)

Ferralium alloy has a design stress of 240 MN/m^2, just below the required 250 MN/m^2. It has excellent corrosion resistance and good machinability, although care is required with screw threads as Ferralium on Ferralium can gall. Ferralium can be surface hardened to 60 RC by Kolsterizing (Ref 116). This process was used on the shaft of the seal rig with no problems.

Under normal working conditions, there is a safety factor of 2 on the thread stresses. Ferralium should therefore be ideal for the inner cylinder, the outer surface, on which the main seals seal, being hardened.

17-4 PH

This is a high strength stainless steel, which has better corrosion resistance than the chromium molybdenum range, although it would not be suitable in severe corrosion conditions. Galling of 17-4 PH thread is a possibility and a suitable thread lubricant should be used.

17-4 PH would be suitable for the outer cylinder, seal cartridge, support column and nozzle holder. Note that it is more expensive than chromium molybdenum.

17-4 PH cannot be hardened to 40 RC and would therefore not be suitable for the inner cylinder.

Inconel Range of Alloys

For high temperature and highly corrosive environments, nickel superalloys, such as the range produced by Inco Alloys International (Ref 117), are the most suitable. They are used regularly in the most severe well conditions for tubulars, valves and other downhole components. Design strengths at 200°C range from 150 MN/m^2 to over 800 MN/m^2 and hardnesses of over 40 RC are possible, depending on the alloy and the heat treatment.

This range of alloys would be suitable for all of the stress carrying manipulator components. However, the material is expensive and would only be used for the most severe well conditions.

Wear Plate Materials

A wear plate is required to cover the nozzle. There are several possible options :

1) Use an ordinary steel which can be through hardened to over 60 RC. Gauge plate was used for the wear plate on the Cutting Head and although only a few tests were conducted, the surface was only polished by the reflected jet.

   This material can readily be machined to the required dimensions before hardening. The only worry, provided that they can be shown to survive 20 hours, is their corrosion resistance. However, as they will be replaced after each operation, corrosion should not be a worry.

2) Aluminium Oxide Ceramic Tile

   A ceramic tile could be moulded and fired to the required shape to protect the nozzle and nozzle holder. These tiles are extremely wear resistant and are also corrosion resistant. The only worry is that, because of the limited space available, the tile could be too thin and brittle to survive the knocks and bangs while the manipulator is being placed in the well. This would need to be checked.

3) Tungsten Carbide Cover

A thin cover plate could be protected by a layer of tungsten carbide. Tungsten carbide would certainly provide the wear resistance, the actual nozzle is generally made of tungsten carbide. It is important that a controlled thickness can be applied, so that the allowable OD of the manipulator is not exceeded. Ideally, a thin layer would be applied to the wear plate in 1 above.

8.5.4 Summary of Material Selection

In general, the following materials should be suitable :

Outer Cylinder	Chromium Molybdenum steel - outer surface protected if required.
Inner Cylinder	Ferralium, surface hardened to 60 RC.
Nozzle Holder	Chromium Molybdenum steel.
Nozzle	Tungsten Carbide.
Nozzle Cap	Chromium Molybdenum.
Wear Plate	Hardened Gauge plate with tungsten carbide layer.
Inner Cylinder Liner	Hardened carbon steel tube.
Seal Cartridge	Chromium Molybdenum.
Support Column	Chromium Molybdenum.
Motor Housing	Chromium Molybdenum.

8.6 Manipulator Assembly

Finally, the assembly of the manipulator needs to be described. The position of the joints in the outer cylinder are shown in Figure 8.6.1 and Appendix G discusses the design and selection of each joint. A detailed description of the assembly sequence is presented below:

1) Clean all components and apply thread lubricant thoroughly.

2) Grease the seals thoroughly and assemble in the seal cartridge and inner cylinder, see Figures 7.9.1 and A1.2.1.

3) Mount the seal cartridge in the outer cylinder and hand tighten the outer cylinder to hold the cartridge firmly in place.

4) Mount the bearing assembly on the support column.

5) Position the support column in the outer cylinder so that the needle bearings are correctly located.

6) Protect the threads of the lower inner cylinder with a plastic cover and gently push the inner cylinder through the seals in the seal cartridge.

7) Assemble the nozzle and wear plate in the nozzle holder and test for leaks around the nozzle (any leaks should be self sealed when abrasive is passed through the nozzle).

8) Position the nozzle holder in the slot in the support column and connect the lower inner cylinder to the nozzle holder by rotating the inner cylinder. Carefully tighten the lower inner cylinder into the nozzle holder and then tighten the grub screws using the access ports through the side of the support column.

9) Connect the motor assembly and lead screw to the inner cylinder.

10) Mount the lead screw nut into the lower outer cylinder and fix in place with the key and locking nuts. Wind the lead screw nut onto the lead screw by turning the lower outer cylinder and connect the lower outer cylinders. It should now be possible to operate the lead screw so that the lower inner cylinder can be moved to the top of the support column.

11) The upper inner cylinder can now be connected to the nozzle holder and tightened.

12) The bottom of the outer cylinder should be plugged and the top connected to a pump. The manipulator should then be pressurised to check for leaks. Leaking connections should be tightened and rechecked. The pump and plug can then be disconnected.

13) The inner cylinder liner can be mounted and the protected motor cable threaded through the complete inner cylinder.

14) Using the ports in the support column, the cavity between the inner cylinder and support column can be filled with a suitable low friction grease. The ports can then be plugged closed.

15) The support column loading pins can now be fixed and the protective cover placed in the slot in the support column.

16) The lower manipulator can be completed, including flushing valve and shear pin system.

17) The upper manipulator can be connected to the additional coiled tubing tools before it is finally ready for connection to the coiled tubing.

A special support stand will be required to enable this assembly.

8.7 Summary

This chapter, together with Appendix G, details the main design considerations for an abrasive water jet window cutting manipulator and shows that the proposed design can be machined and assembled to carry the predicted loads and provide the necessary actuation forces.

Obviously it is only a paper design and considerable refinement will be necessary. However, there is sufficient detail to demonstrate that such a manipulator is feasible.

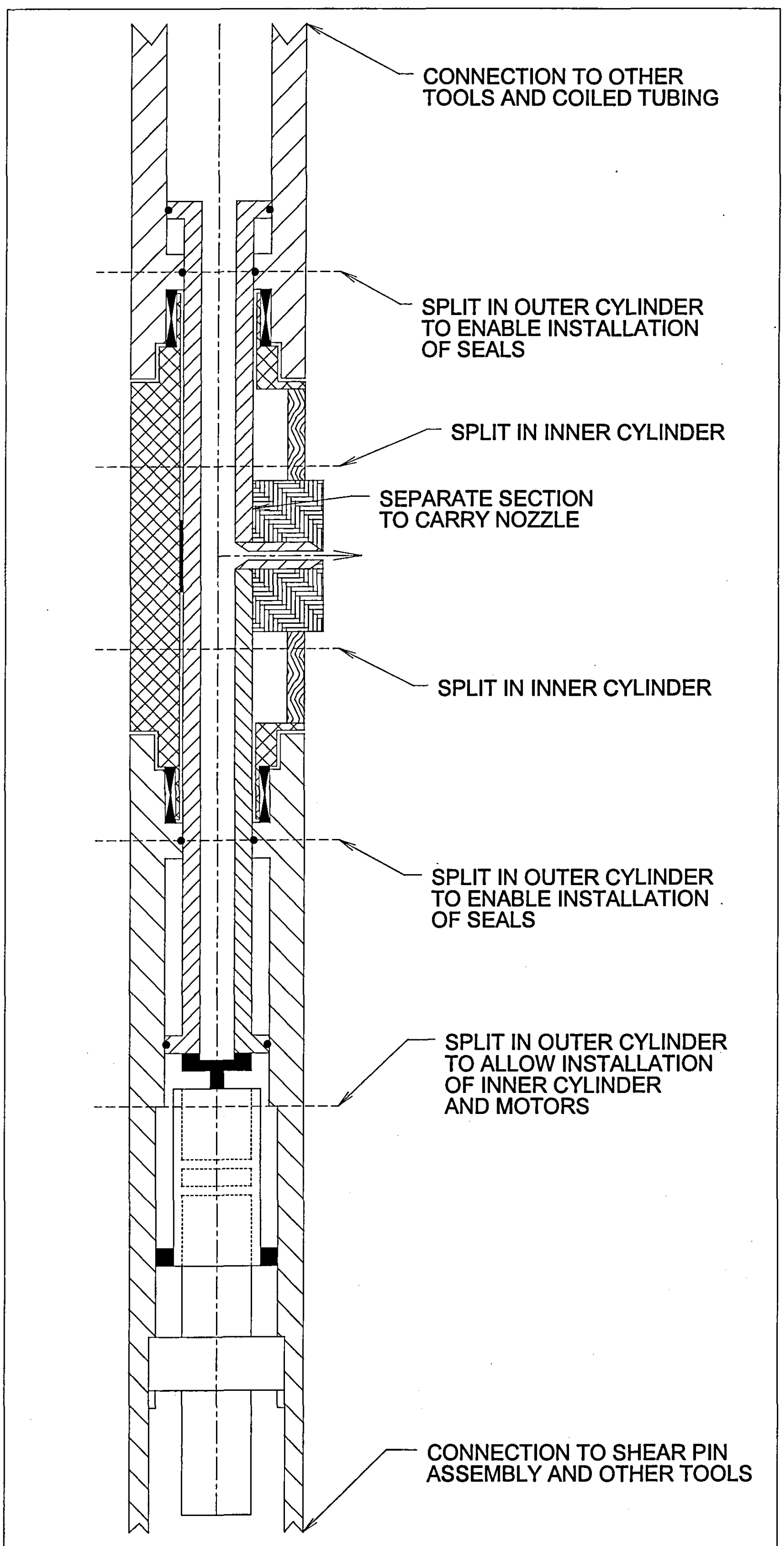


FIGURE 8.6.1 POSITIONS OF SCREW THREAD CONNECTIONS
(See Figure 7.11.1)

9.0 CONCLUSIONS

9.1 Introduction

DIAJET Limited, a subsidiary of BHR Group Limited, manufacture abrasive water jet cutting (AWJC) equipment. They are continually looking for new applications for these machines and cutting inside oil wells was identified as a possible lucrative market. This thesis discusses my investigations into this possibility.

9.2 The Advantages of Abrasive Water Jets

Compared to conventional oil well cutting techniques, abrasive water jets have the great advantage that no force has to be applied to the cutting tool. The jet reaction forces are also relatively low, which means that the nozzle can be manipulated easily, although accurate manipulators are required to ensure a complete cut.

Abrasive water jets can be generated using a direct injection system, such as DIAJET, or an entrainment system. The DIAJET system, which works at lower pressures and only requires one supply hose to the nozzle, is more efficient and has been found to be much more suitable for downhole submerged cutting.

9.3 Possible Downhole Applications

From discussions with representatives from oil companies and oil service companies and a comprehensive literature review, several potential downhole cutting and cleaning applications were identified, together with some points of concern relating to the feasibility and reliability of using an AWJC tool downhole. Of particular concern was the guarantee that all of the abrasive and cuttings could be removed from the well, either during cutting or as a separate operation.

For the first time, each of these concerns has been addressed, either by conducting experiments or from previous experiences. The general concept of using an abrasive water jet (AWJ) downhole is therefore good, provided certain conditions are met. These include:

- using a specially designed cutting fluid with appropriate polymer additives to give excellent solids carrying properties and low pressure loss.

- having a minimum pressure drop across the cutting nozzle of 250 to 300 bar (3600 - 4350 psi). Depending on the depth of operation, this typically corresponds to a minimum surface pressure of 500 to 600 bar (7250 - 8700 psi).

- using the largest possible nozzle to maximise the cutting performance. The stand-off distance should be as small as possible as the cutting efficiency of a submerged jet drops rapidly for stand-off distances greater than 5 nozzle diameters.

- having a manipulator which moves the nozzle smoothly and accurately at variable speeds to ensure a complete cut. Any sudden jumps could result in part of the target being left uncut. This is obviously unacceptable. The manipulator is therefore a critical factor in the feasibility of using an AWJ downhole.

- using large safety factors on the traverse speed because there is no reliable method of monitoring the progress of the cut. Examining the cut after the AWJ has been turned off may be possible.

Of the general applications identified, only a few were sufficiently promising to pursue further, current techniques performing much better than an AWJC tool could achieve. However, there are occasions where these conventional techniques cannot be used in a particular well. An AWJC tool may then be a suitable alternative.

9.4 Through-Tubing Window Cutting

Through-Tubing Window Cutting was found to have the greatest potential. This involves cutting a hole in the oil well casing, below the production tubing, so that a new well can be directionally drilled through it and into the producing formation. Further investigations found that there was a particular requirement for a window cutting tool which could pass through $2\frac{7}{8}$" production tubing and a 2.205" landing nipple; current window cutting techniques have not been developed to achieve this and controlling the weight on such a small diameter cutting bit would be very difficult.

To ensure the window is completely cut and that all of the cuttings are small, the nozzle must be traversed over the complete surface area of the window, in a zig-zag pattern. A model was written to calculate the time required to cut a window using simple Newtonian pressure loss calculations for water and traverse speeds obtained from cutting trials. Large safety factors were included to account for cutting in a submerged environment. The model showed that:

- a 2.5 m by 0.1 m window can be cut in 10 mm thick casing in approximately 12 hours. This assumes that the window is 3000 m away from the surface and that a surface pressure of 595 bar is used. The manipulator would be deployed on 1½" coiled tubing.

- this cutting performance can be achieved with several different nozzles sizes. Ideally a 3 mm diameter nozzle should be used to maximise the cutting performance with stand-off.

- the cutting time should reduce if polymer additives are considered, as they have better pressure loss characteristics, provided the cutting performance is not decreased by using them.

As a guide, a through-tubing window can generally be cut in approximately 10 hours (access through 3½" production tubing), although there is no guarantee of its size and shape. If a suitable manipulator can be designed to control the nozzle, the exact size of window can be cut every time. It is this potential reliability which is particularly attractive, the 12 hours cutting time is then less critical.

A full specification and a detailed design for a window cutting manipulator has therefore been written to demonstrate that window cutting is a feasible application for an AWJ.

9.5 A Through-Tubing Abrasive Water Jet Window Cutting Manipulator

The manipulator basically consists of a hollow inner cylinder, which carries the cutting nozzle, with each end of the cylinder being enclosed and sealed in an outer cylinder. When the system is pressurised, both ends of the inner cylinder are then pressure balanced. To translate or rotate the inner cylinder, and therefore the nozzle, only the break-out frictional forces associated with the seals have to be overcome.

The break-out forces are critical. If they are too high, then too large an actuator will be required to fit in the space available. A seal test rig was designed to measure these forces at pressures up to 600 bar and temperatures up to 175°C. James Walker Limited provided a set of special seals to test. The tests showed that the forces were low, being in the order of 1300 N in translation and 3.5 Nm in rotation.

Suitable permanent magnet electric motors are available to provide several times these forces. The nozzle can therefore be moved accurately by using one motor, coupled to a lead screw, to translate the nozzle in steps of less than the nozzle diameter and a second motor to rotate the nozzle. By using a support column around the inner cylinder, with a bearing assembly at each end which locates in the outer cylinder, the nozzle can be rotated through 360°. A flushing valve is incorporated into the manipulator so that the well can be periodically flushed clean with abrasive-free fluid.

Due to the applied loads, the stroke of the nozzle is limited to approximately 200 mm. To cut longer sections the whole nozzle assembly needs to be moved over a distance of less than the stroke. This is achieved by using a shear pin assembly, with each set of shear pins separated by a distance of less than the stroke. The coiled tubing is held in tension during cutting to maintain its position. At the end of a cut, the tubing is depressurised to below 350 bar and is reeled in to break the shear pin and move the nozzle assembly to the next pin position. The tubing is then repressurised and cutting restarted.

Note that to minimise coiled tubing fatigue, it must not be moved under high internal pressure. The tubing is always stationary during cutting.

9.6 Positioning the Manipulator in the Well

Finally the manipulator must be positioned in the well so that the stand-off distance between the nozzle and the casing is minimal. A conventional through-tubing milling technique is therefore adopted:

> A cement plug is placed in the region where the window needs to be cut and a hole is directionally drilled into the cement towards the casing wall. Once the bit reaches the casing wall, a straight hole, parallel to the wall, is cut. A packer is then inserted at the bottom of this hole, onto which the window cutting manipulator locates. The manipulator is now directly alongside the casing to be cut, minimising the stand-off distance.

Once the window has been cut, the wash-out cavity behind the casing must be filled with cement, to minimise the chance of the drilling assembly getting stuck. The sidetrack well can then be directionally drilled through the cement into the formation.

9.7 Summary

The basic design in conjunction with the test results demonstrates that it is feasible to cut a window using an AWJ. Sufficiently detailed information is given for manufacturing drawings of a prototype to be made.

The pressure balanced design principle is applicable to any AWJC manipulator in which the nozzle is perpendicular to the direction of the well. For example, for perforating short sections of casing or for cutting around the circumference of the production tubing for removal.

10.0 FUTURE WORK

A considerable amount of work still needs to be undertaken before a working tool can be manufactured. The work can basically be divided into two sections:

- the cutting fluid.
- a prototype tool.

10.1 The Cutting Fluid

Potential cutting fluids have been identified but they need to be tested to confirm:

1) their effect on the cutting performance. Generally, polymer additives help to maintain the coherence of the jet and improve the cutting performance.

2) how the polymer additives are to be used with the DIAJET system. The easiest method is to pump them directly through the DIAJET. However, this may make filling the pressure vessel difficult because the abrasive will not settle. The alternative is to run the DIAJET with water as usual and add the polymers after the pressure vessel. This should not be difficult.

3) their ability to suspend the abrasive and cuttings and carry them to the surface. This is the most critical aspect of the whole concept of using AWJ's downhole.

The program to calculate the time to cut a window needs to be modified to account for non-Newtonian fluids and the new cutting performance.

10.2 A Prototype Window Cutting Tool

Several aspects of the design can be built and tested quite easily. In particular, a model of the support column should be made to check the break-out forces for the bearings, and different types of protective covers should be tested to ensure no abrasive can get trapped in the cavity between the inner cylinder and the support column. However, the only way to fully determine that such a tool is going to be reliable is to build a prototype. The prototype, which must include a DIAJET, must be able to demonstrate that it can repeatedly cut the same size of window, and operate continuously for over 20 hours.

DIAJET Limited are currently investigating the different funding options that are available if a prototype tool is to be built. This could be with a single client or as a joint project with several oil companies and oil service companies. Such a joint industry project may be suitable for European Commission funding.

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APPENDIX A CALCULATION OF NOZZLE DIAMETER

The general equation for fluid flow through a nozzle is:

$$\Delta P_N = \frac{1}{2}\rho u^2 = \frac{1}{2}\rho\left(\frac{Q}{A}\right)^2$$

where:

ΔP_N	= pressure drop across the nozzle (N/m^2)
u	= jet velocity leaving the nozzle (m/s)
Q	= volume of cutting fluid (m^3/s)
A	= cross sectional area of the nozzle (m^2) = $\frac{\pi d_N^2}{4}$
d_N	= nozzle diameter (m)
ρ	=fluid density

The actual volume flow, Q_A, is given by:

$$Q_A = Q_{Ideal} * Cd$$

where Cd is the coefficient of discharge for the nozzle which can vary between 0.85 and 0.95 depending on the nozzle (Ref 2).

Substituting A and Cd into the general equation and rearranging, we can obtain an expression for the diameter of the nozzle in terms of the flow rate and the pressure drop across the nozzle:

$$d_N = \left(\frac{4Q}{\pi Cd}\right)^{\frac{1}{2}}\left(\frac{\rho}{2\Delta P_N}\right)^{\frac{1}{4}}$$

Consider, as an example, the following flow:

$\Delta P_N = 350$ bar $\qquad Q = 64.8$ l/min $= 1.08*10^{-3}$ m^3/s

ρ = 1000 kg/ m^3 $\qquad Cd = 0.85$

$$d_N = \left(\frac{4*1.08*10^{-3}}{\pi*0.85}\right)^{\frac{1}{2}}\left(\frac{1000}{2*350*10^5}\right)^{\frac{1}{4}}$$

$$d_N = 2.4*10^{-3}m$$
$$d_N = 2.4mm$$

Table A1 shows a matrix of nozzle sizes at different volumes and pressures for water as the cutting fluid.

TABLE A1 CUTTING FLUID VOLUMES FOR DIFFERENT NOZZLES AND PRESSURES.

	Nozzle Diameter mm							
Pressure bar	0.1	0.2	0.4	0.6	0.8	1.0	1.2	1.4
350	0.1	0.4	1.8	4.0	7.2	11.2	16.2	22.0
700	0.2	0.6	2.5	5.7	10.2	15.9	22.9	31.1
1000	0.2	0.8	3.0	6.8	12.1	19.0	27.3	37.2
1400	0.2	0.9	3.6	8.1	14.4	22.4	32.3	44.0
2000	0.3	1.1	4.3	9.7	17.2	26.8	38.6	52.6
2300	0.3	1.2	4.6	10.4	18.4	28.8	41.4	56.4
	Nozzle Diameter mm							
Pressure bar	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0
350	28.7	36.4	44.9	54.3	64.6	75.9	88.0	101.0
700	40.6	51.4	63.5	76.8	91.4	107.3	124.4	142.8
1000	48.6	61.5	75.9	91.8	109.2	128.2	148.7	170.7
1400	57.5	72.7	89.8	108.6	129.3	151.7	175.9	202.0
2000	68.7	86.9	107.3	129.8	154.5	181.3	210.3	241.4
2300	73.6	93.2	115.1	139.2	165.7	194.5	225.5	258.9

$Cd = 0.9$ $\rho = 1000$ Kg/m^3 Volume in l/min.

APPENDIX B CALCULATION OF AMBIENT BACK PRESSURE REQUIRED TO SUPPRESS CAVITATION

Kolle (Ref 14) explains that both high and low pressure jets are governed by the equations for turbulent jet dissipation. An axisymmetric jet discharged into a fluid of the same density has a centreline velocity, u_m, given by:

$$u_m = u_0 \qquad \textit{for } x < 6.57\, C_d^{\frac{1}{2}}\, d$$

$$u_m = 6.57\, C_d^{\frac{1}{2}}\, d\, \frac{u_0}{x} \qquad \textit{for } x > 6.57\, C_d^{\frac{1}{2}}\, d$$

where u_0 is the jet exit velocity $= \left(\frac{2P}{\rho}\right)^{\frac{1}{2}}$,

and:

x	distance from the nozzle
d	nozzle diameter
C_d	nozzle discharge coefficient
P	nozzle pressure drop
ρ	fluid density.

Note that for a nozzle coefficient of discharge of 0.9, $\frac{x}{d_0} \approx 6$. That is, the centreline velocity should be constant for a distance of 6 nozzle diameters. Belvin (Ref 15) shows that the centreline velocity should be constant for a distance of 5 nozzle diameters away from the nozzle.

If the jet is discharged into low pressure ambient fluid, the dynamic pressure fluctuations in the turbulent mixing region surrounding the jet can cause cavitation.

Cavitation is controlled by the ratio of ambient pressure, P_{amb}, to dynamic jet pressure, which is called the cavitation number, C_{cav} :

$$C_{cav} = \frac{P_{amb}}{P}$$

Cavitation begins to occur for cavitation numbers of about 0.2 or less, and sustained high-intensity cavitation occurs at numbers of less than about 0.1.

APPENDIX C PRESSURES INSIDE THE WELL

C1 The Bottomhole Pressure

The bottomhole pressure is the sum of the hydrostatic pressure exerted by the column of drilling fluids in the well, plus the annular pressure loss of the circulating fluid, plus any surface applied back pressure.

The surface applied back pressure is equal to the drilling fluid pressure leaving the pump minus the pressure losses incurred while moving to the bottom of the well.

For the small volumes of cutting fluid calculated in Section 5.3, the annular pressure loss is small enough to be ignored.

C2 The Formation Fluid Pressure

Sedimentary rock is formed by the accumulation of rock debris and organic material, generally in shallow seas. During this process, water is trapped or invades these formations. As the depth of sediment increases, the rocks are compacted, squeezing the water out. The water within the rocks becomes progressively more salty as the relatively small molecules of water move through the pore spaces of the rock leaving behind the larger salt molecules. The resulting formation fluid pressure is then equivalent to the hydrostatic pressure of a column of salt water.

If the sediment is deposited too quickly for the water to escape, the water begins to bear some of the weight of the sediment, the overburden weight. Pressure within the sediment then increases at a rate equal to the overburden pressure and not just at a rate equal to the pressure of a column of salty water.

An average figure for the normal formation pressure gradient in marine basin sediment in the US Gulf Coast area is 0.465 psi/ft (10.52 kPa/m), although generally this tends to be a little high (Ref 118).

The overburden pressure gradient is normally about 1.0 psi/ft (22.62 kPa/m) (Ref 39).

The density of the drilling mud required to balance the well can then be calculated by assuming an overburden pressure of 1.0 psi/ft:

$$\rho_{mud} = \frac{overburden\ pressure\ gradient}{g}$$

$$= \frac{22.62 * 10^3}{9.81}$$

$$= \underline{2306\ kg/m^3}$$

APPENDIX D MANUFACTURING DRAWINGS FOR THE CUTTING HEAD

The following manufacturing drawings are provided:

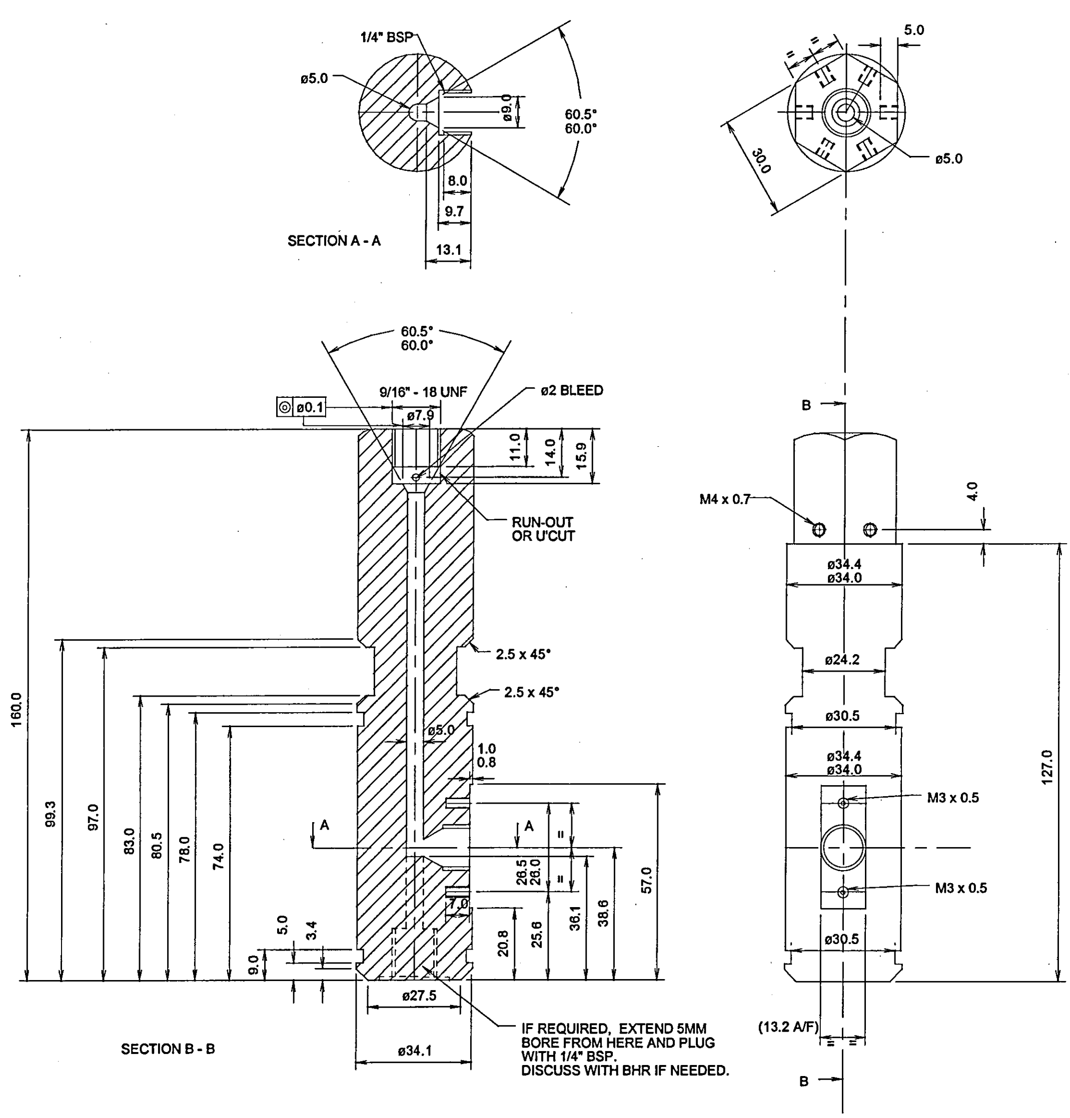
	Drawing Number
Cutting Head	CH02
End Plug	CH03
Support Plate	CH04
Wear Plate	CH05
Nozzle Holder	CH06
Nozzle Holder Adapter	CH07

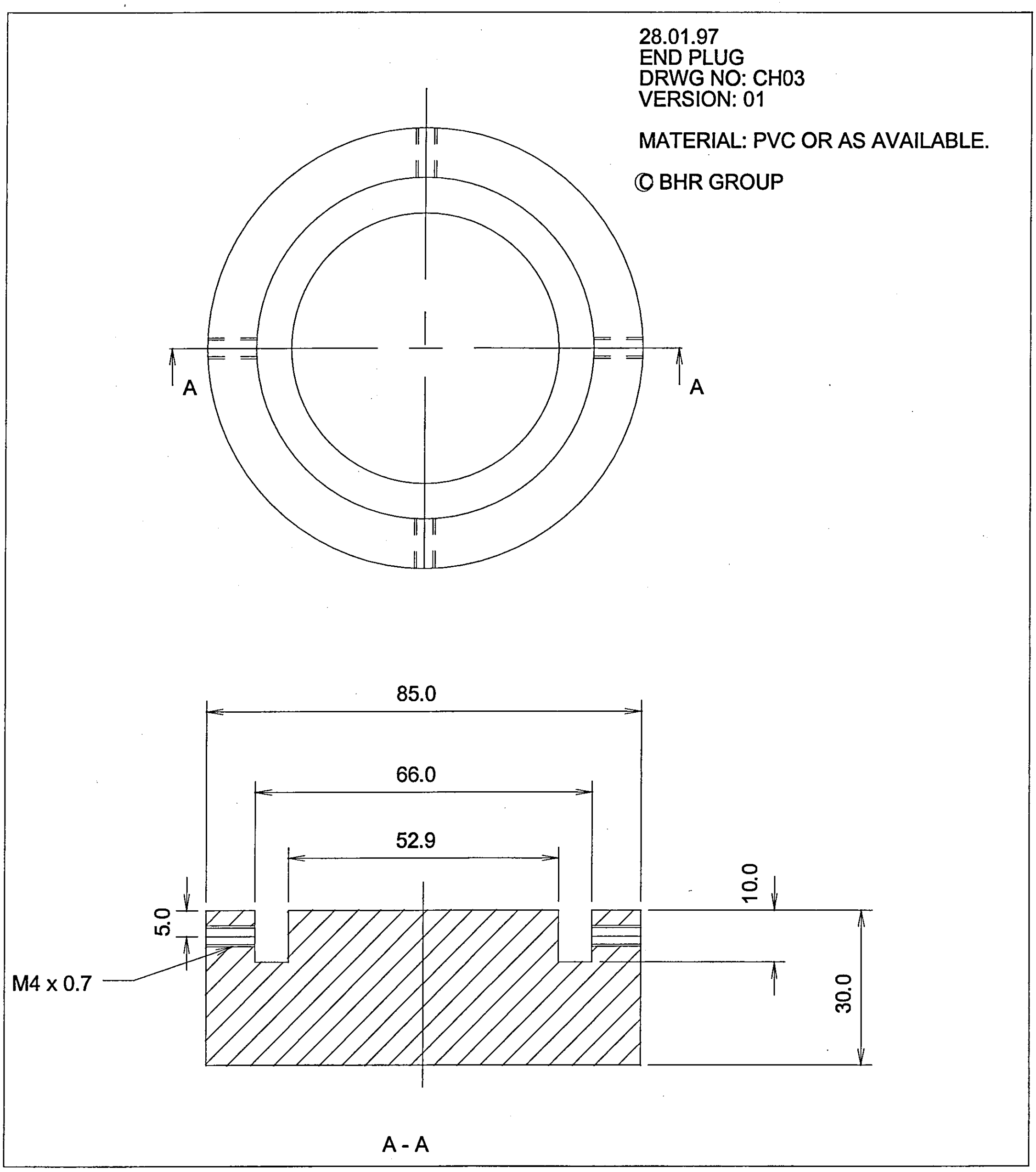
Note that CH04 and CH05 are together on the same page. So are drawings CH06 and CH07.

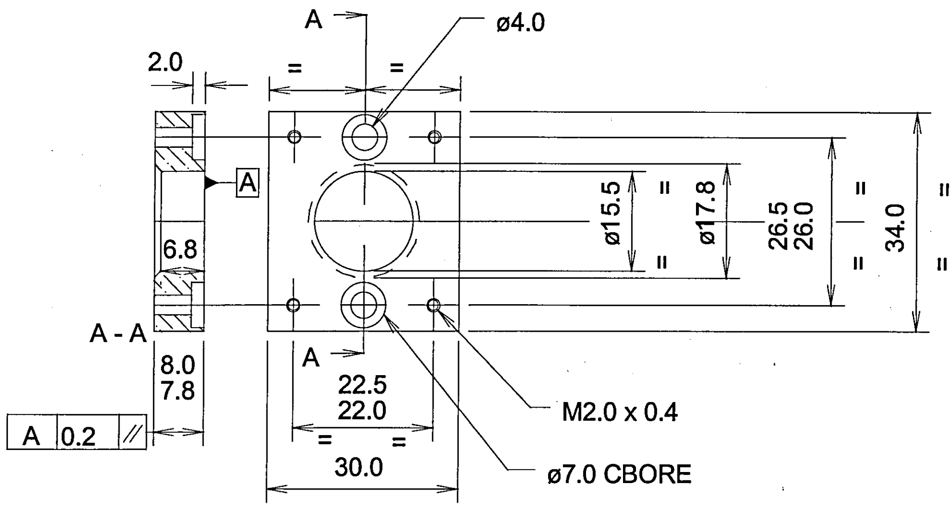
See Figure 7.8.1 for the assembly drawing.

30.01.97
NOZZLE HEAD
DRWG NO: CH02
VERSION: 01

MATERIAL: 316 STAINLESS STEEL

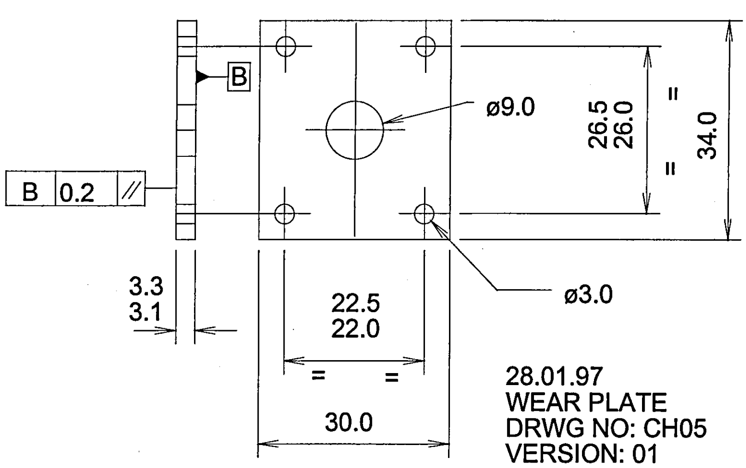



28.01.97
END PLUG
DRWG NO: CH03
VERSION: 01
MATERIAL: PVC OR AS AVAILABLE.
© BHR GROUP
A
A
85.0
66.0
52.9
5.0
10.0
30.0
M4 x 0.7
A - A



SUPPORT PLATE
DRWG NO: CH04
VERSION: 01

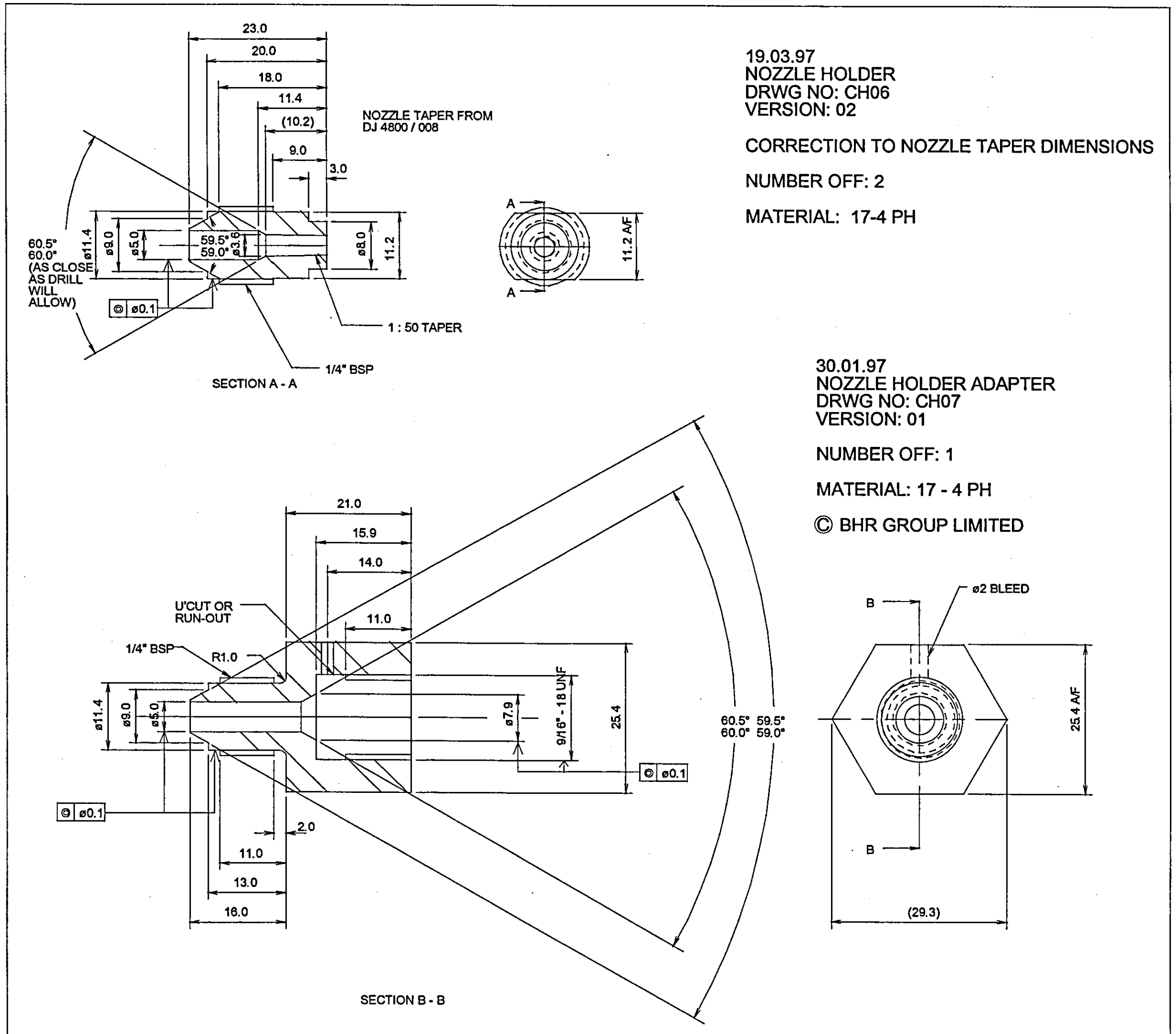
MATERIAL: STAINLESS STEEL (ANY GRADE)
NUMBER OFF: 2



28.01.97
WEAR PLATE
DRWG NO: CH05
VERSION: 01

MATERIAL: GAUGE PLATE
HARDENED TO 60 ROCKWELL C

NUMBER OFF: 8


19.03.97
NOZZLE HOLDER
DRWG NO: CH06
VERSION: 02
CORRECTION TO NOZZLE TAPER DIMENSIONS
NUMBER OFF: 2
MATERIAL: 17-4 PH
NOZZLE TAPER FROM DJ 4800 / 008
23.0
20.0
18.0
11.4
(10.2)
9.0
3.0
60.5° 60.0° (AS CLOSE AS DRILL WILL ALLOW)
ø11.4
ø9.0
ø5.0
59.5° 59.0°
ø3.6
ø8.0
11.2
ø0.1
1 : 50 TAPER
1/4" BSP
SECTION A - A
A
11.2 A/F
30.01.97
NOZZLE HOLDER ADAPTER
DRWG NO: CH07
VERSION: 01
NUMBER OFF: 1
MATERIAL: 17 - 4 PH
© BHR GROUP LIMITED
21.0
15.9
14.0
11.0
U'CUT OR RUN-OUT
1/4" BSP
R1.0
ø11.4
ø9.0
ø5.0
ø7.9
9/16" - 18 UNF
25.4
ø0.1
2.0
11.0
13.0
16.0
60.5° 59.5°
60.0° 59.0°
SECTION B - B
B
ø2 BLEED
25.4 A/F
(29.3)

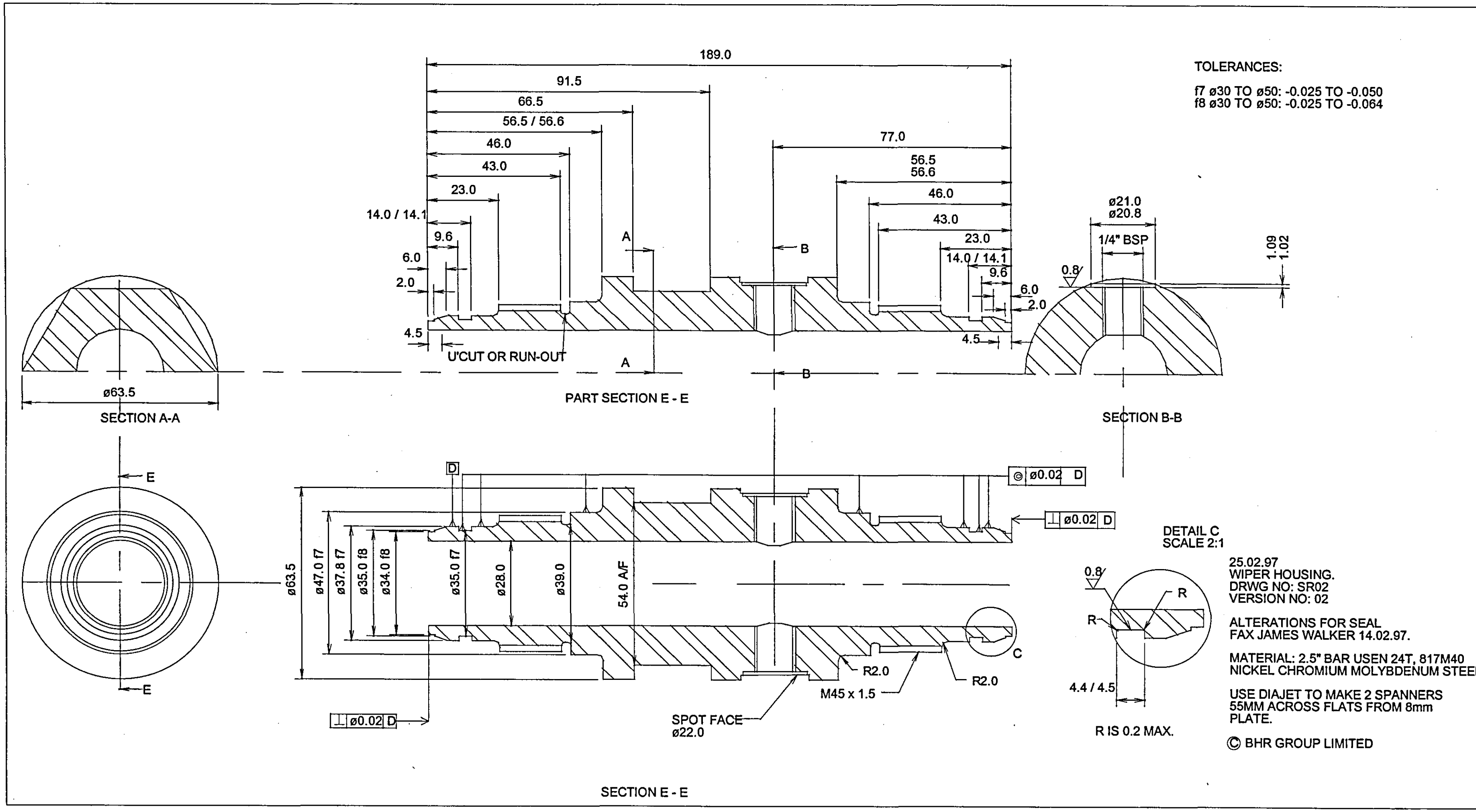
APPENDIX E SEAL RIG

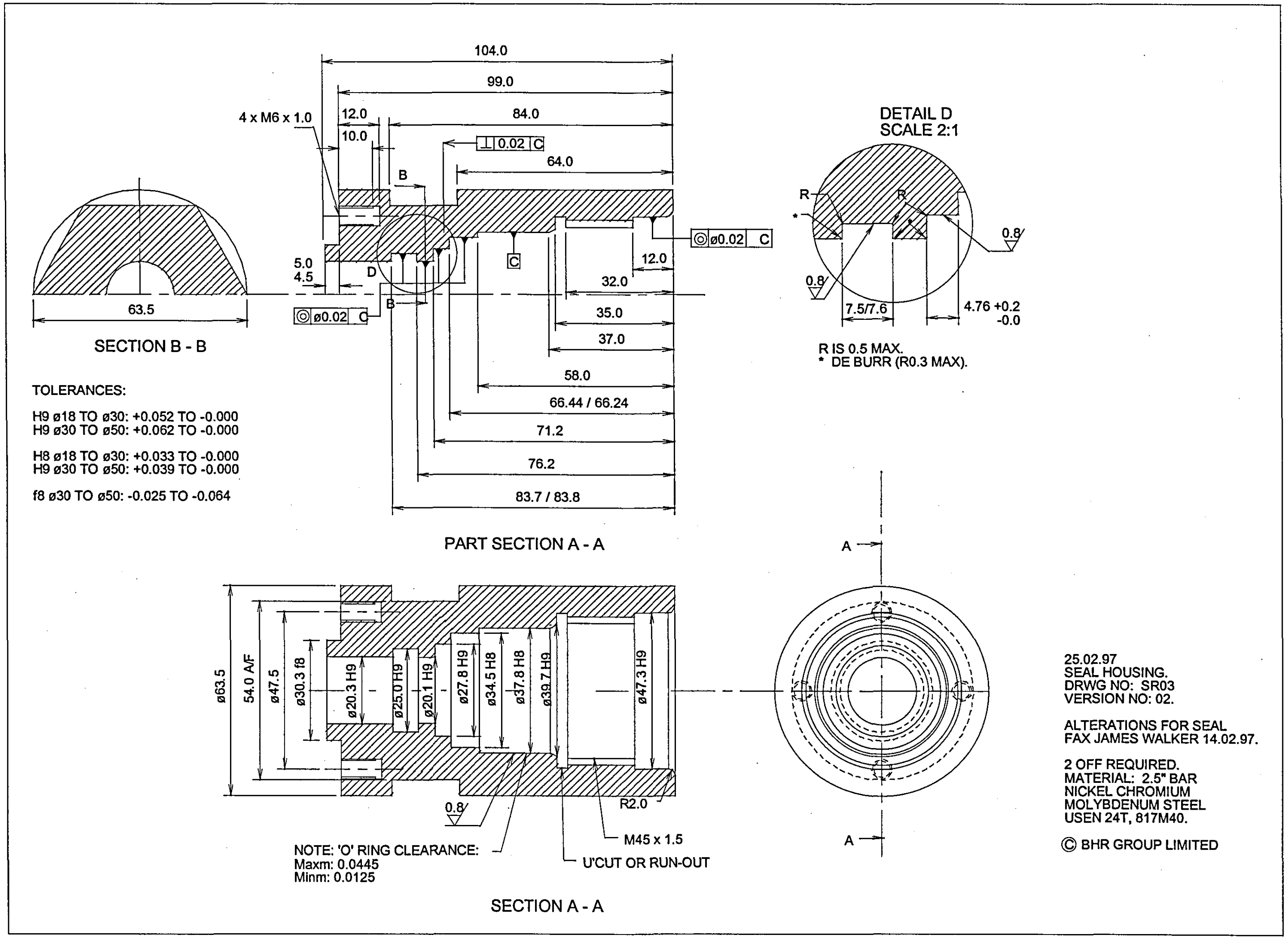
E1 Manufacturing Drawings

The following manufacturing drawings are provided:

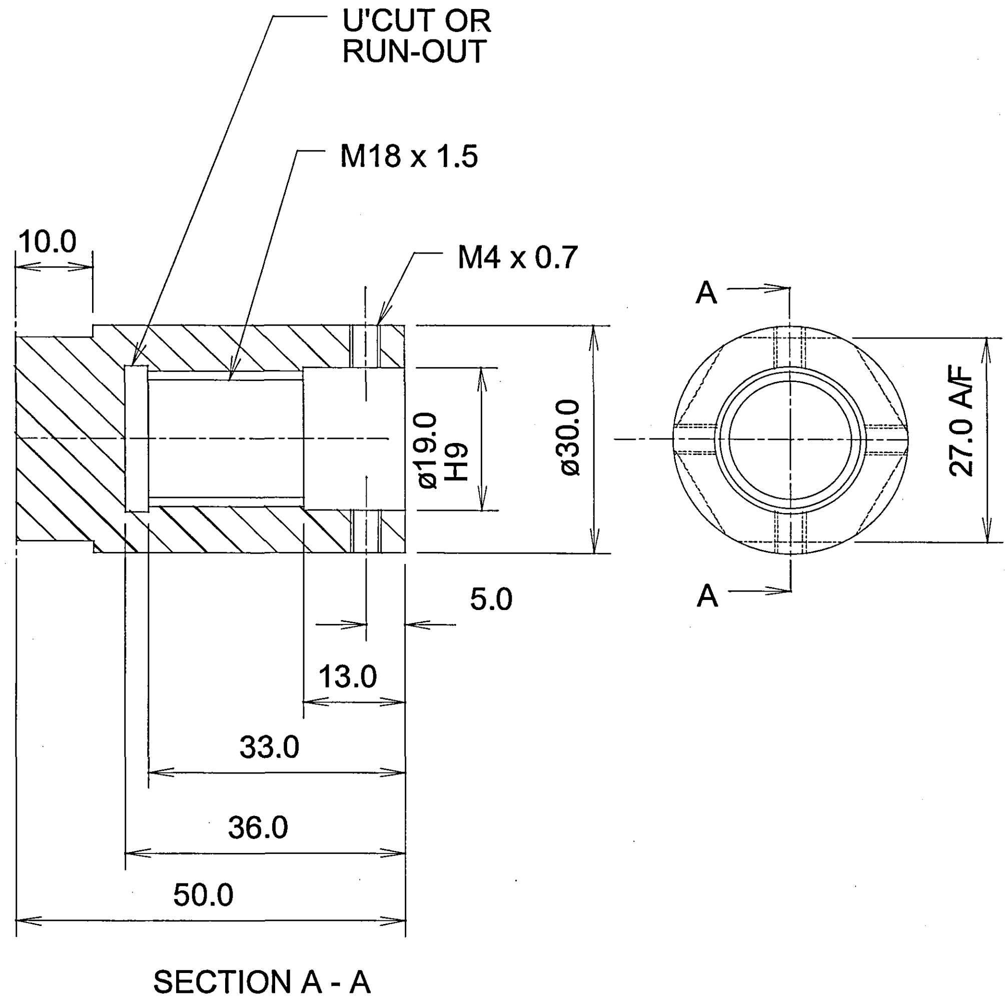
	Drawing Number
Wiper Housing	SR02
Seal Housing	SR03
Shaft Connector	SR04
Face Plate	SR05
Back Up Ring	SR06
Shaft	SR07
Shaft Connector	SR08
Support Plate	SR09
Shaft Thread Cover	SR10

The assembly drawing is shown in Figure 7.10.1.


189.0
91.5
66.5
56.5 / 56.6
46.0
43.0
23.0
14.0 / 14.1
9.6
6.0
2.0
4.5
U'CUT OR RUN-OUT
77.0
56.5
56.6
46.0
43.0
23.0
14.0 / 14.1
9.6
6.0
2.0
4.5
A
B
PART SECTION E - E
ø63.5
SECTION A-A
ø21.0
ø20.8
1/4" BSP
0.8
1.09
1.02
SECTION B-B
TOLERANCES:
f7 ø30 TO ø50: -0.025 TO -0.050
f8 ø30 TO ø50: -0.025 TO -0.064
E
D
ø0.02 D
ø63.5
ø47.0 f7
ø37.8 f7
ø35.0 f8
ø34.0 f8
ø35.0 f7
ø28.0
ø39.0
54.0 A/F
R2.0
M45 x 1.5
R2.0
C
SPOT FACE
ø22.0
SECTION E - E
DETAIL C
SCALE 2:1
0.8
R
R
4.4 / 4.5
R IS 0.2 MAX.
25.02.97
WIPER HOUSING.
DRWG NO: SR02
VERSION NO: 02
ALTERATIONS FOR SEAL
FAX JAMES WALKER 14.02.97.
MATERIAL: 2.5" BAR USEN 24T, 817M40
NICKEL CHROMIUM MOLYBDENUM STEEL.
USE DIAJET TO MAKE 2 SPANNERS
55MM ACROSS FLATS FROM 8mm
PLATE.
© BHR GROUP LIMITED



104.0
99.0
84.0
4 x M6 x 1.0
12.0
10.0
0.02 C
64.0
B
5.0
4.5
D
B
ø0.02 C
C
ø0.02 C
12.0
32.0
35.0
37.0
58.0
66.44 / 66.24
71.2
76.2
83.7 / 83.8
PART SECTION A - A
63.5
SECTION B - B
TOLERANCES:
H9 ø18 TO ø30: +0.052 TO -0.000
H9 ø30 TO ø50: +0.062 TO -0.000
H8 ø18 TO ø30: +0.033 TO -0.000
H9 ø30 TO ø50: +0.039 TO -0.000
f8 ø30 TO ø50: -0.025 TO -0.064
DETAIL D
SCALE 2:1
R
R
0.8
0.8
7.5/7.6
4.76 +0.2 -0.0
R IS 0.5 MAX.
* DE BURR (R0.3 MAX).
ø63.5
54.0 A/F
ø47.5
ø30.3 f8
ø20.3 H9
ø25.0 H9
ø20.1 H9
ø27.8 H9
ø34.5 H8
ø37.8 H8
ø39.7 H9
ø47.3 H9
0.8
R2.0
M45 x 1.5
U'CUT OR RUN-OUT
NOTE: 'O' RING CLEARANCE:
Maxm: 0.0445
Minm: 0.0125
SECTION A - A
A
A
25.02.97
SEAL HOUSING.
DRWG NO: SR03
VERSION NO: 02.
ALTERATIONS FOR SEAL
FAX JAMES WALKER 14.02.97.
2 OFF REQUIRED.
MATERIAL: 2.5" BAR
NICKEL CHROMIUM
MOLYBDENUM STEEL
USEN 24T, 817M40.
© BHR GROUP LIMITED



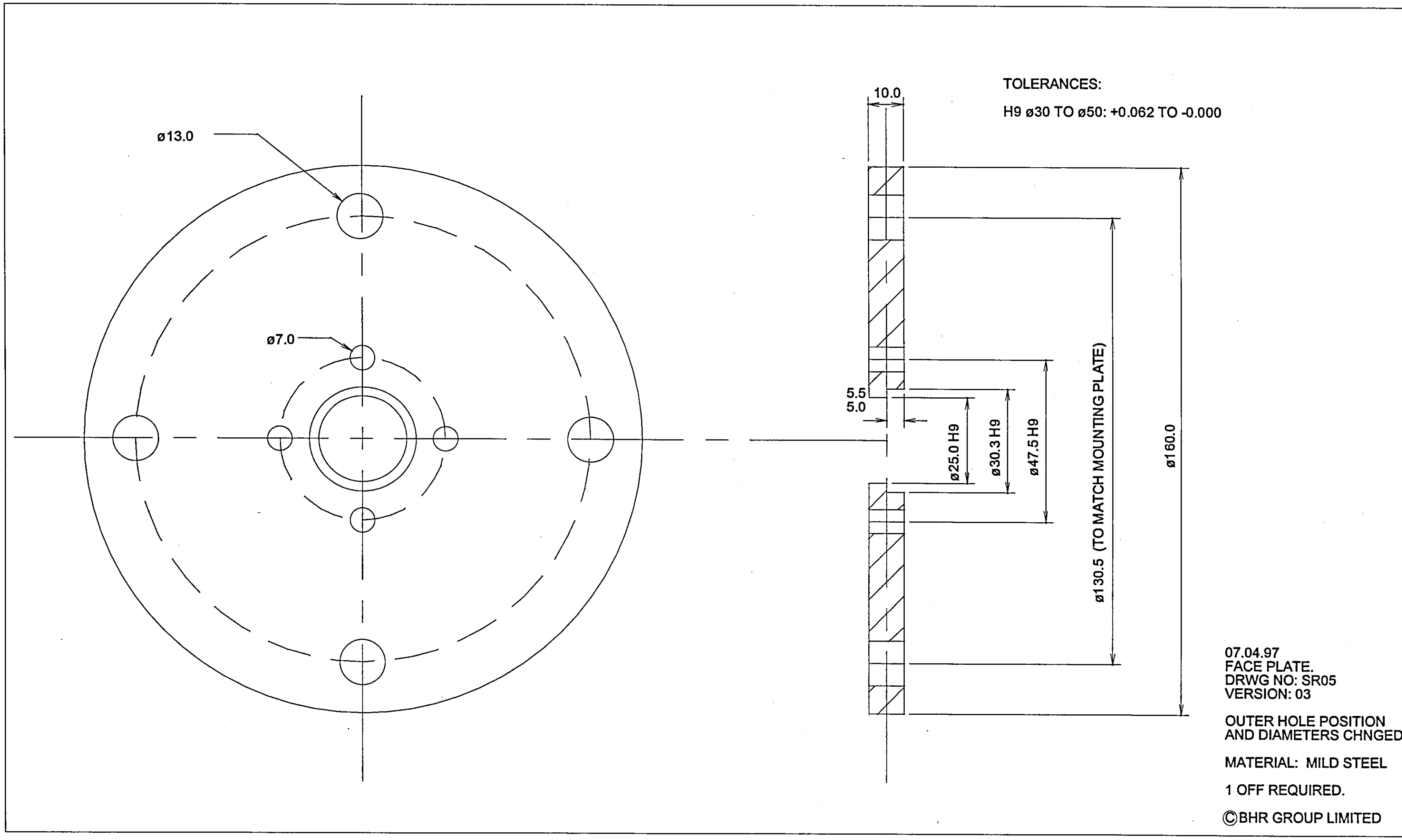
TOLERANCES:

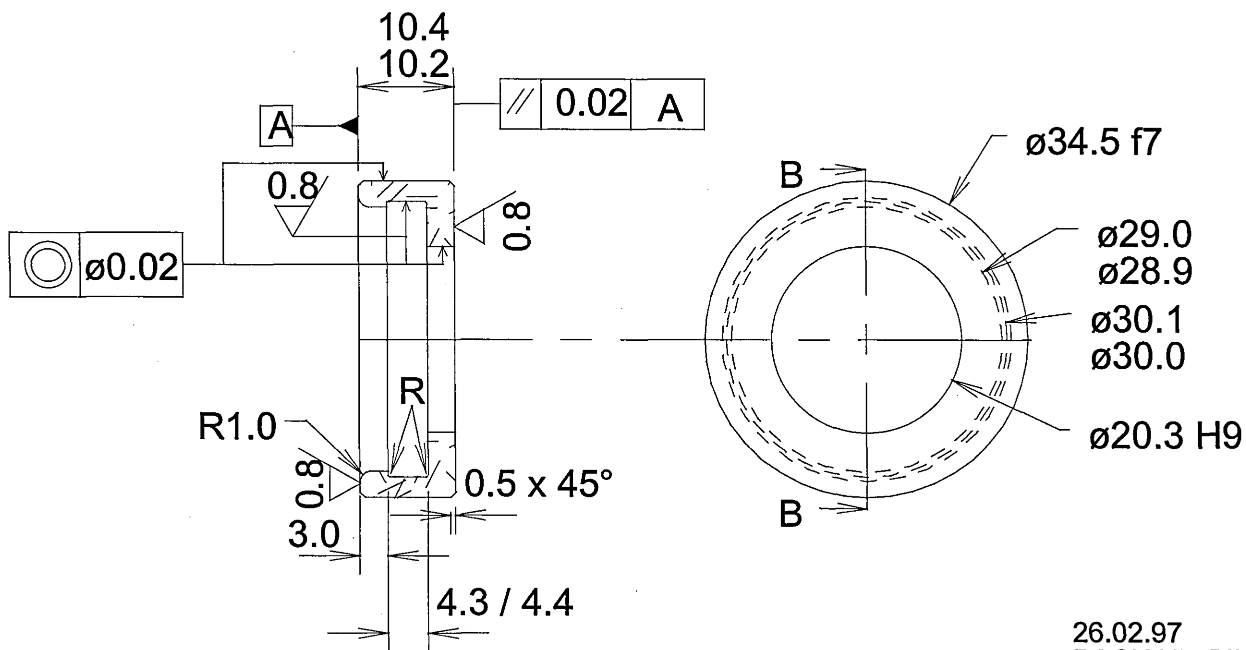
H9 ø18 TO ø30: +0.052 TO -0.000

04.03.97
SHAFT CONNECTOR.
DRWG NO: SR04
VERSION: 02
MATERIAL: MILD STEEL

1 CONNECTORS REQUIRED.

CUT 1 HEX SPANNER 27.2 A/F
USING DIAJET. USE 8mm PLATE.


ø13.0
ø7.0
10.0
5.5
5.0
ø25.0 H9
ø30.3 H9
ø47.5 H9
ø130.5 (TO MATCH MOUNTING PLATE)
ø160.0
TOLERANCES:
H9 ø30 TO ø50: +0.062 TO -0.000
07.04.97
FACE PLATE.
DRWG NO: SR05
VERSION: 03
OUTER HOLE POSITION
AND DIAMETERS CHNGED.
MATERIAL: MILD STEEL
1 OFF REQUIRED.
©BHR GROUP LIMITED



SECTION B-B

R IS 0.5 MAX.

TOLERANCES:
f7 -0.025 TO -0.050
H9 +0.062 TO -0.000

26.02.97
BACK UP RING
DRWG NO: SR06
VERSION: 02

ALTERATIONS FOR SEALS
FAX JAMES WALKER 14.02.97.

MATERIAL 304SS
2 OFF REQUIRED.

© BHR GROUP LIMITED

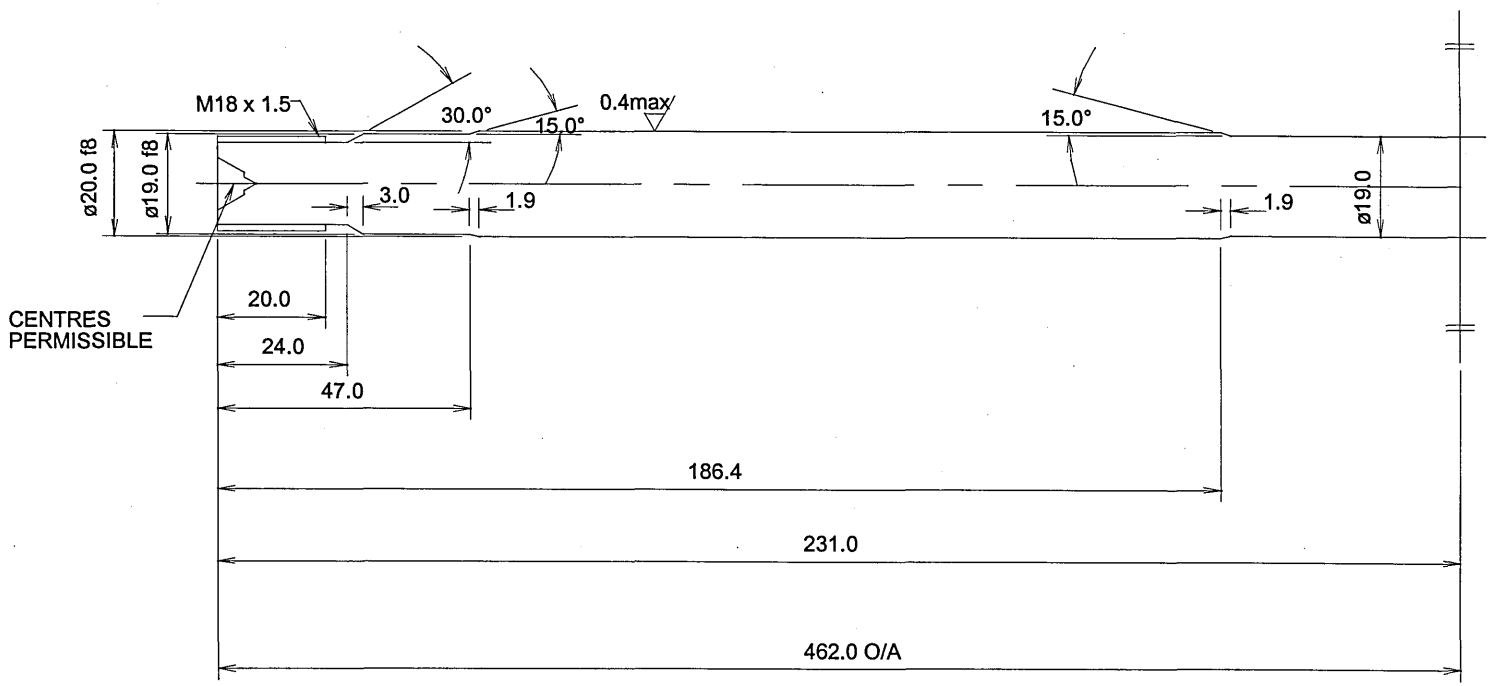
TOLERANCES:

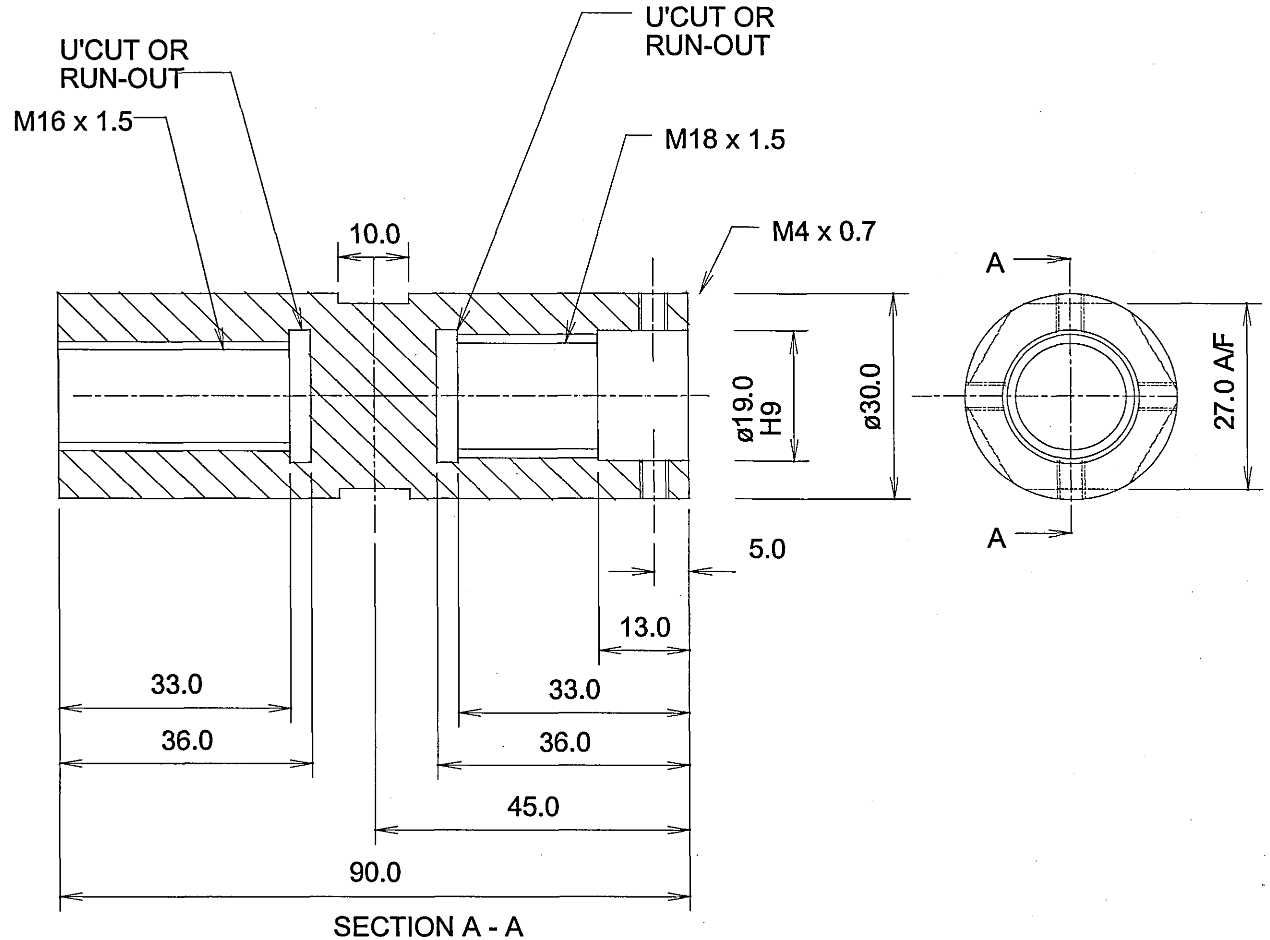
f8 ø18 TO ø30: -0.020 TO -0.053

04.03.97
SHAFT DESIGN
DRWG NO: SR07
VERSION: 02

ALTERATIONS FOR SEAL
FAX JAMES WALKER 14.02.97

MATERIAL:
316SS
FULLY SURFACE TREAT WITH HARDCORE TO 33MICRONS DEPTH.
(1200 - 1500 VICKERS)

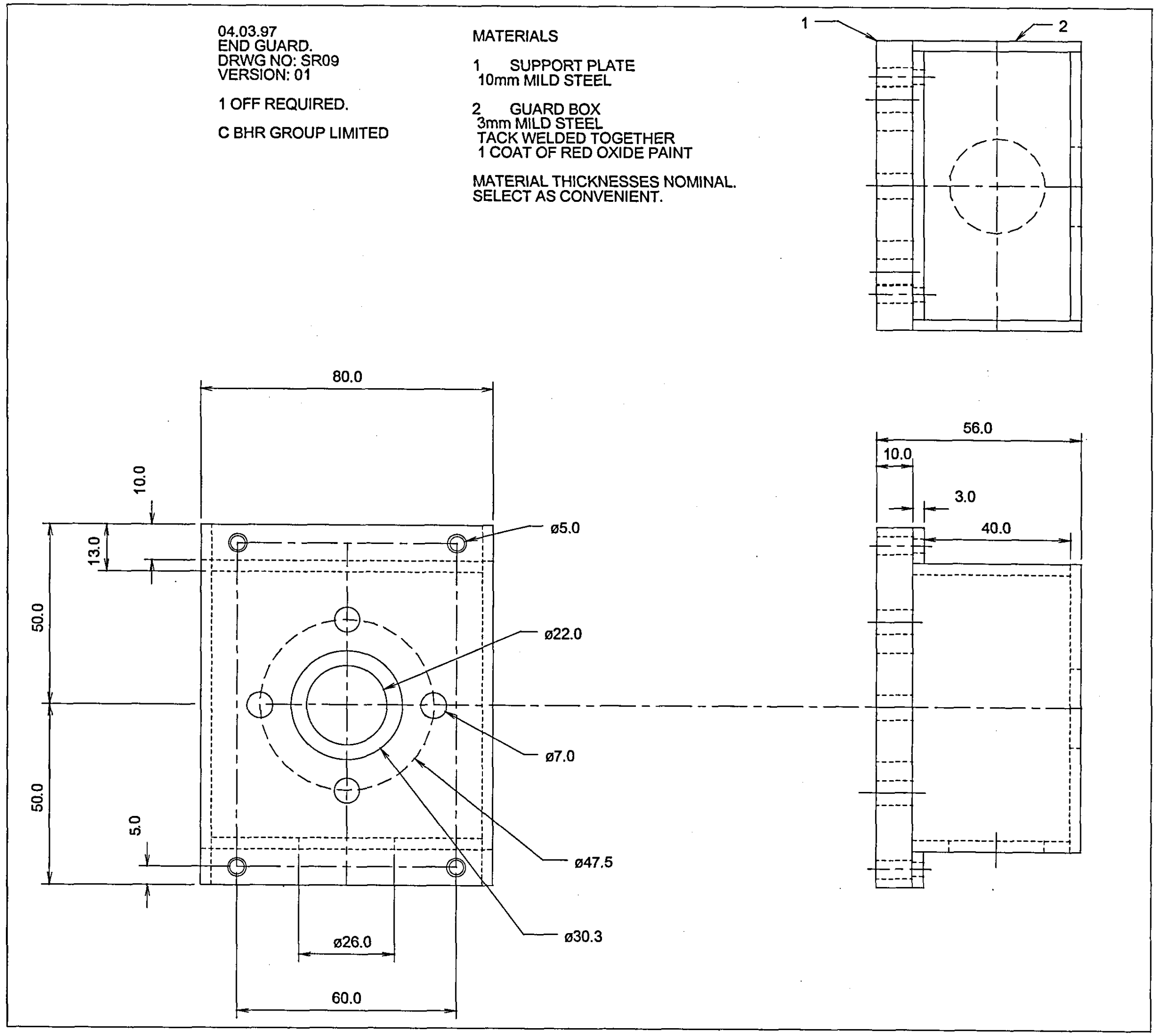


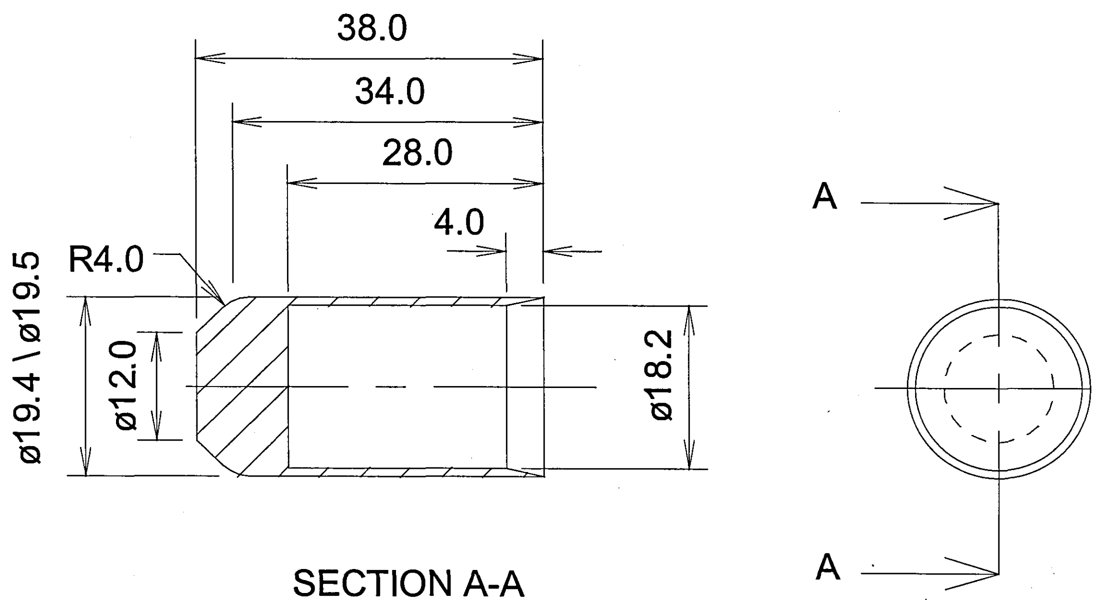


04.03.97
SHAFT CONNECTOR.
DRWG NO: SR08
VERSION: 02
MATERIAL: MILD STEEL

1 CONNECTOR REQUIRED.

CUT 1 HEX SPANNER 27.2 A/F
USING DIAJET. USE 8mm PLATE.


04.03.97
END GUARD.
DRWG NO: SR09
VERSION: 01
1 OFF REQUIRED.
C BHR GROUP LIMITED
MATERIALS
1 SUPPORT PLATE
10mm MILD STEEL
2 GUARD BOX
3mm MILD STEEL
TACK WELDED TOGETHER
1 COAT OF RED OXIDE PAINT
MATERIAL THICKNESSES NOMINAL.
SELECT AS CONVENIENT.
1
2
80.0
10.0
13.0
50.0
50.0
5.0
ø5.0
ø22.0
ø7.0
ø47.5
ø30.3
ø26.0
60.0
56.0
10.0
3.0
40.0



SECTION A-A

26.02.97
SHAFT THREAD COVER
DRWG NO.:SR10
VERSION : 02

CORRECTION TO COVER OD

MATERIAL:
ANY PLASTIC

2 OFF REQUIRED.

E2 Seal Test Results

The next pages show the raw data for the following seal tests:

Test 1	Linear Motion Test at Ambient Temperature and Different Pressures.
Test 2	Rotary Motion Test at Ambient Temperature and Different Pressures.
Test 3	Linear Motion Test at 350 Bar and Different Temperatures.
Test 4	Rotary Motion Test at 350 Bar and Different Temperatures.

The measurements taken for Numbers 1 and 2 for the above four tests were incorrect and were repeated:

Test T1	Linear Motion Test at Ambient Temperature and Different Pressures.
Test T2	Rotary Motion Test at Ambient Temperature and Different Pressures.
Test T3	Linear Motion Test at Ambient Temperature and Different Pressures.
Test T4	Rotary Motion Test at Ambient Temperature and Different Pressures.
Test T5	Linear Motion Test at 350 Bar and Different Temperatures.
Test T6	Rotary Motion Test at 350 Bar and Different Temperatures.
Test T7 (1, 2, and 3)	Linear Motion Test at 600 Bar and Ambient Temperatures for Increasing Number of Cycles.
Test T7 (1, 2, and 3)	Rotary Motion Test at 600 Bar and Ambient Temperatures for Increasing Number of Cycles.

Note that the linear and rotary tests for T7 were measured together.

New Set of Seals:

Test T8	Linear Motion Test at Ambient Temperature and Different Pressures.
Test T9	Rotary Motion Test at Ambient Temperature and Different Pressures.
Test T10	Rotary Motion Test at Ambient Temperature and Different Pressures for a Lighter Lever Arm.

Test 1 Linear Motion Test at Ambient Temperature and Different Pressures.

LINEAR MOTION TEST.

CALIBRATION FACTOR= 2000 [1 VOLT =2000N] TEST NUMBER 1 DATE 3-7-97

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N
1	03:47 PM	0	22	0	0.158	0.038	316	240	0.023	0.023	46	0	0.081	0.022	162	118	0.138	0.035	276	206
2	03:51 PM	2	22	0	0.036	0.036	72	0	0.024	0.024	48	0	0.158	0	316	316	0.123	0.034	246	178
3	03:54 PM	7	22	100	0.036	0.035	72	2	0.003	0	6	6	0.168	-0.022	336	380	0.191	0.056	382	270
4	03:55 PM	8	22	100	0.054	0.054	108	0	-0.004	-0.003	-8	-2	0.236	-0.002	472	476	0.19	0.055	380	270
5	03:58 PM	11	22	200	0.052	0.051	104	2	-0.018	-0.027	-36	18	0.232	-0.016	464	496	0.243	0.066	486	354
6	03:59 PM	12	22	200	0.066	0.062	132	8	-0.01	-0.017	-20	14	0.171	-0.022	342	386	0.24	0.067	480	346
7	04:01 PM	14	22	300	0.067	0.058	134	18	-0.055	-0.069	-110	28	0.264	-0.054	528	636	0.222	0.093	444	258
8	04:03 PM	16	22	300	0.092	0.079	184	26	-0.052	-0.091	-104	78	0.323	-0.033	646	712	0.303	0.097	606	412
9	04:05 PM	18	22	350	0.095	0.081	190	28	-0.072	-0.088	-144	32	0.322	-0.015	644	674	0.304	0.096	608	416
10	04:08 PM	21	22	350	0.115	0.087	230	56	-0.048	-0.05	-96	4	0.309	-0.09	618	798	0.317	0.094	634	446
11	04:11 PM	24	22	400	0.106	0.081	212	50	-0.027	-0.03	-54	6	0.349	-0.068	698	834	0.375	0.124	750	502
12	04:13 PM	26	22	400	0.129	0.1	258	58	-0.051	-0.059	-102	16	0.342	-0.024	684	732	0.367	0.119	734	496
13	04:16 PM	29	22	500	0.125	0.086	250	78	-0.088	-0.09	-176	4	0.384	-0.04	768	848	0.418	0.14	836	556
14	04:18 PM	31	22	500	0.152	0.121	304	62	-0.1	-0.1	-200	0	0.467	-0.041	934	1016	0.42	0.14	840	560
15	04:20 PM	33	22	600	0.155	0.106	310	98	-0.099	-0.1	-198	2	0.465	-0.05	930	1030	0.454	0.14	908	628
16	04:22 PM	34	22	600	0.169	0.132	338	74	-0.05	-0.05	-100	0	0.46	-0.06	920	1040	0.441	0.141	882	600

Test 2 Rotary Motion Test at Ambient Temperature and Different Pressures.

ROTARY MOTION TEST. DATE 4-7-97

CALIBRATION FACTOR= 19.6 [1 VOLT =2Kg]
LEVER ARM (m)= 0.45

TEST NUMBER 2

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm
1	09:35 AM	0	19	0	0.032	0.032	0.282	0.000	0.005	0.003	0.044	0.018	0.034	0.002	0.300	0.282	0.059	0.007	0.520	0.459
2	09:39 AM	4	19	0	0.019	0.022	0.168	-0.026	0.009	0.003	0.079	0.053	0.022	0.009	0.194	0.115	0.063	0.007	0.556	0.494
3	09:42 AM	7	19	100	0.155	0.006	1.367	1.314	0.004	0.001	0.035	0.026	0.088	0.001	0.776	0.767	0.145	0.027	1.279	1.041
4	09:44 AM	9	19	100	0.065	0.007	0.573	0.512	0.001	0.001	0.009	0.000	0.092	0.001	0.811	0.803	0.073	0.031	0.644	0.370
5	09:47 AM	12	19	200	0.143	0.009	1.261	1.182	0.003	0.001	0.026	0.018	0.144	0.001	1.270	1.261	0.183	0.058	1.614	1.103
6	09:49 AM	14	19	200	0.088	0.002	0.776	0.759	0.007	0.001	0.062	0.053	0.131	0.000	1.155	1.155	0.191	0.050	1.685	1.244
7	09:51 AM	16	19	300	0.098	0.062	0.864	0.318	0.000	0.000	0.000	0.000	0.180	0.002	1.588	1.570	0.204	0.082	1.799	1.076
8	09:53 AM	18	19	300	0.114	0.113	1.005	0.009	0.007	0.000	0.062	0.062	0.167	0.005	1.473	1.429	0.211	0.057	1.861	1.358
9	09:59 AM	24	19	350	0.184	0.083	1.623	0.891	0.006	0.002	0.053	0.035	0.175	0.004	1.544	1.508	0.255	0.081	2.249	1.535
10	10:02 AM	27	19	350	0.140	0.140	1.235	0.000	0.003	0.002	0.026	0.009	0.175	0.001	1.544	1.535	0.237	0.083	2.090	1.358
11	10:04 AM	29	19	400	0.135	0.104	1.191	0.273	0.005	0.004	0.044	0.009	0.217	0.002	1.914	1.896	0.275	0.091	2.426	1.623
12	10:06 AM	31	19	400	0.212	0.023	1.870	1.667	-0.110	-0.110	-0.970	0.000	0.104	-0.112	0.917	1.905	0.159	-0.013	1.402	1.517
13	10:08 AM	33	19	500	0.061	0.050	0.538	0.097	-0.112	-0.112	-0.988	0.000	0.136	-0.114	1.200	2.205	0.193	-0.011	1.702	1.799
14	10:10 AM	35	19	500	0.041	0.041	0.362	0.000	-0.110	-0.111	-0.970	0.009	0.105	-0.112	0.926	1.914	0.159	-0.015	1.402	1.535
15	10:13 AM	38	19	600	0.038	-0.031	0.335	0.609	-0.110	-0.110	-0.970	0.000	0.165	-0.112	1.455	2.443	0.233	0.011	2.055	1.958
16	10:14 AM	39	19	600	0.065	0.061	0.573	0.035	-0.111	-0.111	-0.979	0.000	0.167	-0.112	1.473	2.461	0.224	0.003	1.976	1.949

Test 3 Linear Motion Test at 350 Bar and Different Temperatures.

LINEAR MOTION TEST. **DATE** 4-7-97

CALIBRATION FACTOR= 2000 [1 VOLT =2000N] **TEST NUMBER** 3

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N
1	10:19 AM	0	19	0	4.233	0.027	8466	8412	0.025	0.025	50	0	0.128	0.02	256	216	0.134	0.043	268	182
2	10:21 AM	2	19	0	0.044	0.044	88	0	0.025	0.025	50	0	0.084	0.02	168	128	0.137	0.037	274	200
3	10:48 AM	27	175	350	-0.147	0.015	-294	-324	-0.106	-0.122	-212	32	0.399	-0.146	798	1090	0.387	0.12	774	534
4	10:49 AM	28	175	350	0.158	0.115	316	86	-0.08	-0.08	-160	0	0.335	-0.09	670	850	0.329	0.115	658	428
5	10:59 AM	40	150	350	0.128	0.075	256	106	-0.028	-0.03	-56	4	0.237	-0.038	474	550	0.238	0.095	476	286
6	11:01 AM	42	150	350	0.119	0.094	238	50	-0.032	-0.04	-64	16	0.213	-0.036	426	498	0.274	0.095	548	358
7	11:09 AM	50	125	350	0.104	0.073	208	62	-0.012	-0.024	-24	24	0.152	-0.02	304	344	0.25	0.09	500	320
8	11:10 AM	51	125	350	0.097	0.082	194	30	-0.014	-0.026	-28	24	0.148	-0.019	296	334	0.246	0.084	492	324
9	11:22 AM	63	100	350	0.087	0.067	174	40	-0.001	-0.015	-2	28	0.235	-0.007	470	484	0.232	0.081	464	302
10	11:23 AM	64	100	350	0.078	0.069	156	18	-0.003	-0.016	-6	26	0.115	-0.009	230	248	0.165	0.07	330	190
11	11:42 AM	83	75	350	0.072	0.07	144	4	0.002	-0.001	4	6	0.225	-0.001	450	452	0.223	0.065	446	316
12	11:44 AM	85	75	350	0.068	0.063	136	10	0.002	0.001	4	2	0.227	0	454	454	0.155	0.063	310	184
13	12:21 PM	122	50	350	0.065	0.063	130	4	0.01	0.01	20	0	0.149	0.009	298	280	0.143	0.047	286	192
14	12:22 PM	123	50	350	0.051	0.051	102	0	0.01	0.01	20	0	0.196	0.008	392	376	0.199	0.047	398	304
15	01:49 PM	210	30	350	0.053	0.05	106	6	0.014	0.014	28	0	0.168	0.012	336	312	0.177	0.038	354	278
16	01:51 PM	212	30	350	0.04	0.04	80	0	0.014	0.013	28	2	0.192	0.013	384	358	0.124	0.041	248	166

Test 4 Rotary Motion Test at 350 Bar and Different Temperatures.

ROTARY MOTION TEST.

DATE 9-7-97

CALIBRATION FACTOR= 19.62 [1 VOLT =2Kg]

LEVER ARM (m)= 0.45

TEST NUMBER 4

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm
1	11:13 AM	0	23	0	-0.003	-0.006	-0.026	0.026	-0.005	-0.006	-0.044	0.009	-0.004	-0.007	-0.035	0.026	0.000	-0.005	0.000	0.044
2	11:16 AM	3	23	0	-0.002	-0.004	-0.018	0.018	0.019	-0.005	0.168	0.212	0.017	-0.005	0.150	0.194	0.007	-0.005	0.062	0.106
3	11:36 AM	23	175	350	0.008	-0.007	0.071	0.132	0.001	-0.020	0.009	0.185	0.267	-0.020	2.357	2.534	0.335	0.214	2.958	1.068
4	11:38 AM	25	175	350	0.249	0.299	2.198	-0.441	-0.014	-0.016	-0.124	0.018	0.218	-0.017	1.925	2.075	0.265	0.189	2.340	0.671
5	11:48 AM	35	150	350	0.180	0.052	1.589	1.130	-0.011	-0.013	-0.097	0.018	0.057	-0.012	0.503	0.609	0.105	0.025	0.927	0.706
6	11:50 AM	37	150	350	0.081	0.072	0.715	0.079	-0.011	-0.011	-0.097	0.000	0.040	-0.013	0.353	0.468	0.099	0.023	0.874	0.671
7	11:57 AM	44	125	350	0.086	0.014	0.759	0.636	-0.011	-0.011	-0.097	0.000	0.029	-0.012	0.256	0.362	0.071	0.000	0.627	0.627
8	11:59 AM	46	125	350	0.056	0.052	0.494	0.035	-0.010	-0.010	-0.088	0.000	0.023	-0.012	0.203	0.309	0.097	0.003	0.856	0.830
9	12:10 PM	57	100	350	0.058	-0.004	0.512	0.547	-0.008	-0.010	-0.071	0.018	0.046	-0.011	0.406	0.503	0.091	0.000	0.803	0.803
10	12:12 PM	59	100	350	0.060	0.056	0.530	0.035	-0.009	-0.010	-0.079	0.009	0.040	-0.012	0.353	0.459	0.095	0.000	0.839	0.839
11	12:30 PM	77	75	350	0.068	0.003	0.600	0.574	0.000	-0.001	0.000	0.009	0.046	0.000	0.406	0.406	0.106	0.004	0.936	0.901
12	12:32 PM	79	75	350	0.048	0.047	0.424	0.009	0.026	0.000	0.230	0.230	0.030	0.000	0.265	0.265	0.107	0.006	0.945	0.892
13	01:08 PM	115	50	350	0.069	0.000	0.609	0.609	-0.002	-0.004	-0.018	0.018	0.067	-0.006	0.592	0.645	0.104	0.008	0.918	0.848
14	01:09 PM	116	50	350	0.062	0.062	0.547	0.000	-0.001	-0.004	-0.009	0.026	0.075	-0.005	0.662	0.706	0.147	0.022	1.298	1.104
15	02:00 PM	167	32	350	0.099	0.024	0.874	0.662	-0.003	-0.003	-0.026	0.000	0.110	-0.005	0.971	1.015	0.182	0.058	1.607	1.095
16	02:02 PM	169	32	350	0.111	0.108	0.980	0.026	-0.002	-0.003	-0.018	0.009	0.091	-0.004	0.803	0.839	0.155	0.042	1.368	0.998

Test T1 Linear Motion Test at Ambient Temperatures and Different Pressures.

LINEAR MOTION TEST.

CALIBRATION FACTOR= 2000 [1 VOLT =2000N] TEST NUMBER T1 DATE 6.8.97

Test Information					Number 1 out				Number 2 out				Number 3 in				Number 4 in			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N
1	10.25 AM	0	17	0	-0.016	0.000	-32	-32	-0.010	0.006	-20	-32	0.048	0.002	96	92	0.046	0.020	92	52
2	10.29 AM	4	18	0	-0.014	0.029	-28	-86	-0.012	0.003	-24	-30	0.048	0.000	96	96	0.047	0.035	94	24
3	10.34AM	9	19	100	-0.111	0.037	-222	-296	-0.072	-0.030	-144	-84	0.179	-0.036	358	430	0.170	0.090	340	160
4	10.36AM	11	19	100	-0.098	0.060	-196	-316	-0.070	-0.034	-140	-72	0.178	-0.029	356	414	0.164	0.060	328	208
5	10.41AM	16	20	200	-0.143	0.049	-286	-384	-0.099	-0.051	-198	-96	0.232	-0.041	464	546	0.221	0.079	442	284
6	10.44AM	19	20	200	-0.132	0.070	-264	-404	-0.108	-0.048	-216	-120	0.238	-0.041	476	558	0.226	0.099	452	254
7	10.46AM	21	20	300	-0.167	0.051	-334	-436	-0.123	-0.054	-246	-138	0.269	-0.047	538	632	0.247	0.100	494	294
8	10.48AM	23	20	300	-0.142	0.080	-284	-444	-0.119	-0.050	-238	-138	0.265	-0.043	530	616	0.246	0.100	492	292
9	10.52AM	27	21	350	-0.200	0.042	-400	-484	-0.123	-0.060	-246	-126	0.291	-0.053	582	688	0.265	0.100	530	330
10	10.54AM	29	21	350	-0.175	0.090	-350	-530	-0.134	-0.052	-268	-164	0.268	-0.047	536	630	0.271	0.100	542	342
11	10.56AM	31	21	400	-0.174	0.061	-348	-470	-0.119	-0.052	-238	-134	0.288	-0.050	576	676	0.277	0.100	554	354
12	10.59AM	34	21	400	-0.178	0.090	-356	-536	-0.115	-0.045	-230	-140	0.308	-0.047	616	710	0.283	0.100	566	366
13	11.01AM	36	21	500	-0.218	0.044	-436	-524	-0.144	-0.060	-288	-168	0.327	-0.056	654	766	0.285	0.100	570	370
14	11.04AM	39	21	500	-0.210	0.090	-420	-600	-0.124	-0.045	-248	-158	0.325	-0.049	650	748	0.289	0.100	578	378
15	11.07AM	42	21	600	-0.279	0.039	-558	-636	-0.153	-0.064	-306	-178	0.374	-0.062	748	872	0.320	0.100	640	440
16	11.10AM	45	21	600	-0.223	0.070	-446	-586	-0.141	-0.057	-282	-168	0.372	-0.056	744	856	0.338	0.100	676	476

Note: At 1 to 2 bar, water began to leak from one seal.

Test T2 Rotary Motion Test at Ambient Temperatures and Different Pressures.

ROTARY MOTION TEST. DATE 6.8.97

CALIBRATION FACTOR= 19.600 [1 VOLT =2Kg]
LEVER ARM (m)= 0.450

TEST NUMBER T2

Test Information					Number 1 out				Number 2 out				Number 3 in				Number 4 in			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm
1	11.56 AM	28	21	0	-0.004	-0.002	-0.035	-0.018	-0.005	-0.005	-0.044	0.000	0.000	-0.002	0.000	0.018	0.003	-0.003	0.026	0.053
2	11.58 AM	30	21	0	-0.001	0.000	-0.009	-0.009	-0.004	-0.001	-0.035	-0.026	0.000	-0.004	0.000	0.035	0.002	0.000	0.018	0.018
3	11.28 AM	0	21	100	-0.010	-0.007	-0.088	-0.026	-0.011	-0.010	-0.097	-0.009	0.000	-0.009	0.000	0.079	0.002	-0.004	0.018	0.053
4	1130 AM	2	21	100	-0.008	-0.004	-0.071	-0.035	-0.007	-0.007	-0.062	0.000	0.000	-0.008	0.000	0.071	0.019	-0.007	0.168	0.229
5	11.32 AM	4	21	200	-0.009	0.007	-0.079	-0.141	-0.010	-0.009	-0.088	-0.009	-0.003	-0.010	-0.026	0.062	0.044	-0.008	0.388	0.459
6	11.34 AM	6	22	200	-0.011	0.003	-0.097	-0.123	-0.010	-0.010	-0.088	0.000	0.016	-0.010	0.141	0.229	0.028	-0.008	0.247	0.318
7	11.36 AM	8	22	300	-0.008	-0.006	-0.071	-0.018	-0.008	-0.007	-0.071	-0.009	-0.001	-0.008	-0.009	0.062	-0.005	-0.008	-0.044	0.026
8	11.39 AM	11	22	300	-0.008	-0.006	-0.071	-0.018	-0.008	-0.008	-0.071	0.000	-0.001	-0.008	-0.009	0.062	0.067	-0.005	0.591	0.635
9	11.40 AM	12	22	350	-0.011	0.004	-0.097	-0.132	-0.011	-0.009	-0.097	-0.018	-0.001	-0.010	-0.009	0.079	0.019	-0.008	0.168	0.238
10	11.42 AM	14	22	350	-0.007	-0.001	-0.062	-0.053	-0.007	-0.007	-0.062	0.000	0.001	-0.007	0.018	0.079	0.099	-0.005	0.873	0.917
11	11.45 AM	17	22	400	-0.008	0.039	-0.071	-0.415	-0.010	-0.008	-0.088	-0.018	0.002	-0.008	0.009	0.079	0.063	-0.005	0.556	0.600
12	11.46 AM	18	22	400	-0.008	0.029	-0.071	-0.326	-0.010	-0.008	-0.088	-0.018	0.001	-0.009	0.009	0.088	0.088	-0.007	0.776	0.838
13	11.49 AM	21	22	500	-0.006	0.034	-0.053	-0.353	-0.006	-0.006	-0.053	0.000	0.001	-0.006	0.035	0.088	0.017	-0.005	0.150	0.194
14	11.51 AM	23	22	500	-0.008	0.001	-0.071	-0.079	-0.010	-0.008	-0.088	-0.018	0.004	-0.011	-0.009	0.088	0.105	-0.008	0.926	0.997
15	11.53 AM	25	22	600	-0.006	0.038	-0.053	-0.388	-0.007	-0.004	-0.062	-0.026	-0.001	-0.006	0.044	0.097	0.047	-0.005	0.415	0.459
16	11.54 AM	26	22	600	-0.006	0.034	-0.053	-0.353	-0.008	-0.006	-0.071	-0.018	0.005	-0.007	0.044	0.106	0.084	-0.004	0.741	0.776

Test T3 Linear Motion Test at Ambient Temperatures and Different Pressures.

LINEAR MOTION TEST.

CALIBRATION FACTOR= 2000 [1 VOLT =2000N] TEST NUMBER T3 DATE 8.8.97

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N
1	17.05	80	24	0	-0.015	0.023	-30	-76	-0.015	0.000	-30	-30	0.039	0.000	78	78	0.036	0.022	72	28
2	17.03	78	24	0	-0.017	0.007	-34	-48	-0.017	-0.004	-34	-26	0.040	-0.007	80	94	0.037	0.021	74	32
3	16.56	71	24	100	-0.064	0.061	-128	-250	-0.067	-0.007	-134	-120	0.170	-0.007	340	354	0.168	0.050	336	236
4	16.54	69	24	100	-0.109	0.039	-218	-296	-0.068	-0.024	-136	-88	0.172	-0.020	344	384	0.166	0.062	332	208
5	16.46	61	24	200	-0.128	0.052	-256	-360	-0.107	-0.026	-214	-162	0.238	-0.034	476	544	0.240	0.080	480	320
6	16.44	59	24	200	-0.143	0.040	-286	-366	-0.094	-0.036	-188	-116	0.247	-0.037	494	568	0.241	0.062	482	358
7	16.38	53	24	300	-0.149	0.060	-298	-418	-0.123	-0.034	-246	-178	0.262	-0.040	524	604	0.264	0.087	528	354
8	16.34	49	24	300	-0.179	0.041	-358	-440	-0.107	-0.040	-214	-134	0.254	-0.030	508	568	0.258	0.064	516	388
9	16.21	36	24	350	-0.166	0.038	-332	-408			0	0	0.273	-0.045	546	636	0.266	0.090	532	352
10	16.19	34	24	350	-0.180	0.034	-360	-428	-0.129	-0.040	-258	-178	0.287	-0.044	574	662	0.275	0.057	550	436
11	16.10	25	24	400	-0.191	0.047	-382	-476	-0.126	-0.043	-252	-166	0.266	-0.047	532	626	0.277	0.090	554	374
12	16.08	23	24	400	-0.199	0.035	-398	-468	-0.138	-0.049	-276	-178	0.303	-0.045	606	696	0.290	0.074	580	432
13	15.57	12	24	500	-0.186	0.060	-372	-492	-0.144	-0.050	-288	-188	0.319	-0.050	638	738	0.319	0.100	638	438
14	15.54	9	24	500	-0.242	0.045	-484	-574	-0.142	-0.032	-284	-220	0.342	-0.040	684	764	0.322	0.080	644	484
15	15.46	1	24	600	-0.218	0.060	-436	-556	-0.163	-0.056	-326	-214	0.345	-0.050	690	790	0.329	0.100	658	458
16	15.45	0	24	600	-0.264	0.038	-528	-604	-0.164	-0.054	-328	-220	0.329	-0.050	658	758	0.337	0.069	674	536

Note: At 1 to 2 bar, water began to leak from one seal.

Test T4 Rotary Motion Test at Ambient Temperatures and Different Pressures.

ROTARY MOTION TEST. DATE 8.8.97

CALIBRATION FACTOR= 19.600 [1 VOLT =2Kg]
LEVER ARM (m)= 0.450

TEST NUMBER T4

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm
1	17.09	81	24	0	0.012	0.012	0.106	0.000	0.011	0.011	0.097	0.000	0.019	0.012	0.168	0.062	0.014	0.011	0.123	0.026
2	17.07	79	24	0	0.014	0.014	0.123	0.000	0.012	0.012	0.106	0.000	0.026	0.014	0.229	0.106	0.017	0.013	0.150	0.035
3	17.00	72	24	100	0.014	0.051	0.123	-0.326	0.012	0.014	0.106	-0.018	0.025	0.012	0.221	0.115	0.066	0.013	0.582	0.467
4	16.59	70	24	100	0.011	0.011	0.097	0.000	0.012	0.012	0.106	0.000	0.023	0.013	0.203	0.088	0.062	0.018	0.547	0.388
5	16.51	62	24	200	0.013	0.061	0.115	-0.423	0.013	0.014	0.115	-0.009	0.016	0.014	0.141	0.018	0.064	0.014	0.564	0.441
6	16.49	60	24	200	0.011	0.012	0.097	-0.009	0.013	0.013	0.115	0.000	0.022	0.013	0.194	0.079	0.070	0.012	0.617	0.512
7	16.41	52	24	300	0.013	0.013	0.115	0.000	0.010	0.013	0.088	-0.026	0.019	0.010	0.168	0.079	0.014	0.012	0.123	0.018
8	16.39	50	24	300	0.012	0.014	0.106	-0.018	0.011	0.013	0.097	-0.018	0.017	0.011	0.150	0.053	0.016	0.015	0.141	0.009
9	16.28	39	24	350	0.013	0.013	0.115	0.000	0.010	0.012	0.088	-0.018	0.022	0.012	0.194	0.088	0.016	0.014	0.141	0.018
10	16.25	36	24	350	0.013	0.013	0.115	0.000	0.012	0.013	0.106	-0.009	0.015	0.011	0.141	0.044	0.025	0.013	0.221	0.106
11	16.16	27	24	400	0.012	0.013	0.106	-0.009	0.013	0.013	0.115	0.000	0.016	0.013	0.194	0.079	0.020	0.012	0.176	0.071
12	16.13	24	24	400	0.000	0.013	0.000	-0.115	0.013	0.013	0.115	0.000	0.022	0.013	0.132	0.018	0.015	0.013	0.132	0.018
13	16.01	12	24	500	0.012	0.014	0.106	-0.018	0.009	0.010	0.079	-0.009	0.015	0.009	0.115	0.035	0.015	0.012	0.132	0.026
14	15.59	10	24	500	0.008	0.011	0.071	-0.026	0.009	0.010	0.079	-0.009	0.013	0.006	0.326	0.256	0.014	0.013	0.123	0.009
15	15.51	2	24	600	0.016	0.172	0.141	-1.376	0.015	0.016	0.132	-0.009	0.037	0.016	0.335	0.194	0.301	0.016	2.655	2.514
16	15.49	0	24	600	0.015	0.015	0.132	0.000	0.013	0.016	0.115	-0.026	0.038	0.013	0.335	0.221	0.221	0.016	1.949	1.808

Test T5 Linear Motion Test at 350 Bar and Different Temperatures.

LINEAR MOTION TEST.

CALIBRATION FACTOR= 2000 [1 VOLT =2000N] TEST NUMBER T5 DATE 11.8.97

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N
1	8.52	0	21	0	-0.017	0.008	-34	-50	-0.029	-0.005	-58	-48	0.050	0.003	100	94	0.051	0.025	102	52
2	8.54	2	22	0	-0.014	0.024	-28	-76	-0.026	0.000	-52	-52	0.051	0.000	102	102	0.052	0.024	104	56
3	9.21	29	175	350	-0.450	-0.019	-900	-862	-0.309	-0.190	-618	-238	0.385	-0.183	770	1136	0.341	0.120	682	442
4	9.22	30	169	350	-0.332	0.090	-664	-844	-0.265	-0.165	-530	-200	0.313	-0.145	626	916	0.321	0.125	642	392
5	9.29	37	150	350	-0.248	0.053	-496	-602	-0.184	-0.114	-368	-140	0.287	-0.090	574	754	0.275	0.097	550	356
6	9.31	39	146	350	-0.236	0.072	-472	-616	-0.182	-0.102	-364	-160	0.263	-0.083	526	692	0.255	0.117	510	276
7	9.39	47	125	350	-0.229	0.045	-458	-548	-0.152	-0.095	-304	-114	0.290	-0.085	580	750	0.257	0.100	514	314
8	9.41	49	122	350	-0.190	0.068	-380	-516	-0.160	-0.092	-320	-136	0.275	-0.082	550	714	0.245	0.116	490	258
9	9.54	62	100	350	-0.171	0.049	-342	-440	-0.132	-0.075	-264	-114	0.280	-0.065	560	690	0.228	0.089	456	278
10	9.55	63	98	350	-0.175	0.058	-350	-466	-0.145	-0.076	-290	-138	0.267	-0.071	534	676	0.235	0.109	470	252
11	10.14	82	75	350	-0.174	0.046	-348	-440	-0.118	-0.067	-236	-102	0.259	-0.054	518	626	0.224	0.080	448	288
12	10.15	83	74	350	-0.162	0.064	-324	-452	-0.146	-0.067	-292	-158	0.253	-0.065	506	636	0.227	0.100	454	254
13	10.54	122	50	350	-0.172	0.043	-344	-430	-0.122	-0.057	-244	-130	0.236	-0.043	472	558	0.225	0.078	450	294
14	10.56	124	51	350	-0.141	0.060	-282	-402	-0.158	-0.054	-316	-208	0.221	-0.052	442	546	0.223	0.094	446	258
15	14.29	337	35	350	-0.163	0.036	-326	-398	-0.134	-0.042	-268	-184	0.196	-0.040	392	472	0.199	0.061	398	276
16	14.30	338	35	350	-0.139	0.050	-278	-378	-0.142	-0.046	-284	-192	0.213	-0.047	426	520	0.202	0.056	404	292

Note: At 1 to 2 bar, water began to leak from one seal.

Test T6 Rotary Motion Test at 350 Bar and Different Temperatures.

ROTARY MOTION TEST. DATE 11.8.97

CALIBRATION FACTOR= 19.600 [1 VOLT =2Kg] TEST NUMBER T6
LEVER ARM (m)= 0.450

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm
1	14.43	0	34	0	0.014	0.016	0.123	-0.018	0.016	0.016	0.141	0.000	0.044	0.017	0.388	0.238	0.040	0.016	0.353	0.212
2	14.46	3	34	0	0.015	0.017	0.132	-0.018	0.014	0.015	0.123	-0.009	0.035	0.015	0.309	0.176	0.027	0.016	0.238	0.097
3	15.10	27	175	350	0.017	0.018	0.150	-0.009	0.017	0.017	0.150	0.000	0.026	0.018	0.229	0.071	0.042	0.018	0.370	0.212
4	15.11	28	173	350	0.017	0.017	0.150	0.000	0.015	0.017	0.132	-0.018	0.022	0.017	0.194	0.044	0.027	0.017	0.238	0.088
5	15.24	41	150	350	0.016	0.017	0.141	-0.009	0.015	0.016	0.132	-0.009	0.024	0.015	0.212	0.079	0.025	0.016	0.221	0.079
6	15.25	42	148	350	0.016	0.017	0.141	-0.009	0.015	0.016	0.132	-0.009	0.025	0.016	0.221	0.079	0.031	0.017	0.273	0.123
7	15.34	51	125	350	0.014	0.016	0.123	-0.018	0.013	0.014	0.115	-0.009	0.029	0.013	0.256	0.141	0.043	0.015	0.379	0.247
8	15.36	53	122	350	0.016	0.016	0.141	0.000	0.016	0.017	0.115	-0.035	0.032	0.016	0.282	0.141	0.029	0.018	0.256	0.097
9	15.49	66	100	350	0.016	0.017	0.141	-0.009	0.016	0.017	0.141	-0.009	0.033	0.016	0.291	0.150	0.027	0.017	0.238	0.088
10	15.50	67	97	350	0.016	0.016	0.141	0.000	0.016	0.017	0.141	-0.009	0.027	0.016	0.335	0.194	0.028	0.017	0.247	0.097
11	16.10	87	75	350	0.017	0.017	0.150	0.000	0.017	0.018	0.141	-0.018	0.038	0.015	0.309	0.176	0.039	0.017	0.344	0.194
12	16.11	88	74	350	0.014	0.016	0.123	-0.018	0.015	0.015	0.150	0.018	0.035	0.015	0.256	0.123	0.038	0.017	0.335	0.185
13	16.54	131	50	350	0.012	0.014	0.106	-0.018	0.011	0.013	0.132	0.018	0.029	0.011	0.379	0.282	0.027	0.013	0.238	0.123
14	16.56	135	50	350	0.013	0.013	0.115	0.000	0.012	0.014	0.097	-0.026	0.043	0.012	0.415	0.309	0.042	0.013	0.370	0.256
15	18.06	205	39	350	0.012	0.013	0.106	-0.009	0.011	0.013	0.106	-0.009	0.047	0.011	0.362	0.265	0.028	0.013	0.247	0.132
16	18.07	206	39	350	0.014	0.016	0.123	-0.018	0.013	0.014	0.115	-0.009	0.041	0.013	0.362	0.247	0.035	0.013	0.309	0.194

Test T71 Linear Motion Test at 600 Bar at Ambient Temperatures for Increasing Number of Cycles.

LINEAR MOTION TEST.

CALIBRATION FACTOR= 2000 [1 VOLT =2000N] TEST NUMBER T71 DATE 12.8.97

Test Information					Number 1 out				Number 4 in				
Stroke	Time when measurements taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Number of Cycles
1	17.21	0	26	600	-0.241	0.046	-482	-574	0.319	-0.017	638	672	50
2	17.21	0	26	600	-0.271	0.042	-542	-626	0.301	-0.005	602	612	50
3	17.29	8	26	600	-0.297	0.007	-594	-608	0.347	-0.072	694	838	50
4	17.30	9	26	600	-0.204	0.049	-408	-506	0.328	-0.043	656	742	50
5	17.32	11	26	600	-0.198	0.062	-396	-520	0.310	0.002	620	616	100
6	17.34	13	26	600	-0.170	0.041	-340	-422	0.302	0.002	604	600	100
7	17.41	20	26	600	-0.232	0.030	-464	-524	0.343	-0.058	686	802	100
8	17.42	22	26	600	-0.215	0.042	-430	-514	0.364	-0.018	728	764	100
9	17.45	25	26	600	-0.161	0.058	-322	-438	0.326	0.002	652	648	150
10	17.46	26	26	600	-0.174	0.073	-348	-494	0.312	0.002	624	620	150
11	17.51	31	26	600	-0.238	0.034	-476	-544	0.372	-0.050	744	844	150
12	17.52	32	26	600	-0.221	0.045	-442	-532	0.337	-0.008	674	690	150
13	17.54	34	26	600	-0.221	0.057	-442	-556	0.323	0.002	646	642	200
14	17.55	35	26	600	-0.188	0.067	-376	-510	0.305	0.002	610	606	200
15	18.01	41	26	600	-0.262	0.032	-524	-588	0.342	-0.020	684	724	200
16	18.01	41	26	600	-0.198	0.043	-396	-482	0.322	0.010	644	624	200

Note: At 1 to 2 bar, water began to leak from one seal.

Test T72 Linear Motion Test at 600 Bar at Ambient Temperatures for Increasing Number of Cycles.

LINEAR MOTION TEST.

CALIBRATION FACTOR= 2000 [1 VOLT =2000N] TEST NUMBER T72 DATE 12 & 13.8.97

Test Information					Number 1 out				Number 4 in				
Stroke	Time when measurements taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Number of Cycles
1	18.04	0	26	600	-0.209	0.045	-418	-508	0.318	0.002	636	632	250
2	18.04	0	26	600	-0.193	0.042	-386	-470	0.321	-0.008	642	658	250
3	18.11	7	26	600	-0.252	0.036	-504	-576	0.372	-0.070	744	884	250
4	18.12	8	26	600	-0.228	0.080	-456	-616	0.336	-0.026	672	724	250
5	18.14	10	26	600	-0.197	0.070	-394	-534	0.270	0.002	540	536	300
6	18.15	11	26	600	-0.228	0.051	-456	-558	0.321	0.002	642	638	300
7	18.22	18	26	600	-0.271	0.029	-542	-600	0.326	-0.026	652	704	300
8	18.23	19	26	600	-0.232	0.091	-464	-646	0.327	-0.010	654	674	300
9	16.53	19	24	600	-0.186	0.087	-372	-546	0.306	-0.010	612	632	350
10	16.54	21	24	600	-0.235	0.044	-470	-558	0.319	-0.010	638	658	350
11	17.00	27	24	600	-0.278	0.041	-556	-638	0.348	-0.076	696	848	350
12	17.01	28	24	600	-0.221	0.103	-442	-648	0.346	-0.045	692	782	350
13	17.04	31	24	600	-0.221	0.063	-442	-568	0.334	-0.011	668	690	400
14	17.04	31	24	600	-0.179	0.053	-358	-464	0.311	-0.012	622	646	400
15	17.11	38	24	600	-0.260	0.039	-520	-598	0.374	-0.069	748	886	400
16	17.12	39	24	600	-0.226	0.096	-452	-644	0.346	-0.020	692	732	400

Note: At 1 to 2 bar, water began to leak from one seal.

Test T73 Linear Motion Test at 600 Bar at Ambient Temperatures for Increasing Number of Cycles.

LINEAR MOTION TEST.

CALIBRATION FACTOR= 2000 [1 VOLT =2000N] TEST NUMBER T73 DATE 13.8.97

Test Information					Number 1 out				Number 4 in				
Stroke	**Time when measurements taken.**	**Elapsed time**	**Temp**	**Press. Bar**	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Number of Cycles
1	17.17	0	25	600	-0.209	0.061	-418	-540	0.327	-0.040	654	734	600
2	17.17	0	25	600	-0.218	0.068	-436	-572	0.327	-0.021	654	696	600
3	17.27	10	25	600	-0.322	0.045	-644	-734	0.357	-0.071	714	856	600
4	17.27	10	25	600	-0.197	0.108	-394	-610	0.347	-0.047	694	788	600
5	17.34	17	25	600	-0.238	0.073	-476	-622	0.344	-0.004	688	696	800
6	17.34	17	25	600	-0.259	0.075	-518	-668	0.345	-0.008	690	706	800
7	17.44	27	25	600	-0.342	0.011	-684	-706	0.310	-0.058	620	736	800
8	17.44	27	25	600	-0.201	0.131	-402	-664	0.340	-0.016	680	712	800
9							0	0			0	0	
10							0	0			0	0	
11							0	0			0	0	
12							0	0			0	0	
13							0	0			0	0	
14							0	0			0	0	
15							0	0			0	0	
16							0	0			0	0	

Note: At 1 to 2 bar, water began to leak from one seal.

Test T71 Rotary Motion Test at 600 Bar at Ambient Temperatures for Increasing Number of Cycles.

ROTARY MOTION TEST. DATE 12.8.97

CALIBRATION FACTOR= 19.600 [1 VOLT =2Kg]
LEVER ARM (m)= 0.450

TEST NUMBER T71

Test Information					Number 1 out				Number 4 in				
Stroke	Time when measurements taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Number of Cycles
1	17.24	0	26	600	0.017	0.024	0.150	-0.062	0.046	0.017	0.406	0.256	50
2	17.25	1	26	600	0.017	0.024	0.150	-0.062	0.065	0.018	0.573	0.415	50
3	17.28	4	26	600	0.024	0.024	0.212	0.000	0.032	0.025	0.282	0.062	50
4	17.28	4	26	600	0.024	0.024	0.212	0.000	0.034	0.025	0.300	0.079	50
5	17.35	11	26	600	0.019	0.023	0.168	-0.035	0.114	0.019	1.005	0.838	100
6	17.36	12	26	600	0.018	0.104	0.159	-0.759	0.125	0.019	1.103	0.935	100
7	17.39	15	26	600	0.019	0.023	0.168	-0.035	0.027	0.023	0.238	0.035	100
8	17.40	16	26	600	0.023	0.023	0.203	0.000	0.028	0.024	0.247	0.035	100
9	17.46	22	26	600	0.023	0.025	0.203	-0.018	0.071	0.023	0.626	0.423	150
10	17.47	23	26	600	0.024	0.065	0.000	-0.362	0.071	0.024	0.626	0.415	150
11	17.49	25	26	600	0.024	0.070	0.212	-0.406	0.072	0.025	0.635	0.415	150
12	17.50	26	26	600	0.025	0.069	0.221	-0.388	0.067	0.025	0.591	0.370	150
13	17.56	32	26	600	0.023	0.075	0.203	-0.459	0.240	0.023	2.117	1.914	200
14	17.57	33	26	600	0.024	0.160	0.212	-1.200	0.175	0.024	1.544	1.332	200
15	17.59	35	26	600	0.024	0.024	0.212	0.000	0.029	0.023	0.256	0.053	200
16	18.00	36	26	600	0.022	0.023	0.194	-0.009	0.026	0.022	0.229	0.035	200

Test T72 Rotary Motion Test at 600 Bar at Ambient Temperatures for Increasing Number of Cycles.

ROTARY MOTION TEST. **DATE** **12 & 13.8.97**

CALIBRATION FACTOR= **19.600** **[1 VOLT =2Kg]**
LEVER ARM (m)= **0.450**

TEST NUMBER T72

Test Information					**Number 1 out**				**Number 4 in**				
Stroke	**Time when measurements taken.**	**Elapsed time**	**Temp**	**Press. Bar**	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Number of Cycles
1	18.06	0	26	600	0.022	0.023	0.194	-0.009	0.324	0.025	2.858	2.637	250
2	18.07	1	26	600	0.023	0.227	0.203	-1.799	0.266	0.023	2.346	2.143	250
3	18.09	3	26	600	0.025	0.025	0.221	0.000	0.042	0.025	0.370	0.150	250
4	18.10	4	26	600	0.025	0.026	0.221	-0.009	0.035	0.026	0.309	0.079	250
5	18.17	11	26	600	0.020	0.042	0.176	-0.194	0.104	0.021	0.917	0.732	300
6	18.18	12	26	600	0.025	0.100	0.221	-0.662	0.105	0.025	0.926	0.706	300
7	18.20	14	26	600	0.026	0.109	0.229	-0.732	0.103	0.026	0.908	0.679	300
8	18.21	15	26	600	0.026	0.093	0.229	-0.591	0.100	0.026	0.882	0.653	300
9	16.56	15	24	600	0.009	0.191	0.079	-1.605	0.356	0.011	3.140	3.043	350
10	16.57	16	24	600	0.009	0.260	0.000	-2.214	0.346	0.010	3.052	2.964	350
11	16.59	18	24	600	0.009	0.234	0.079	-1.985	0.269	0.011	2.373	2.276	350
12	16.59	18	24	600	0.010	0.231	0.088	-1.949	0.267	0.010	2.355	2.267	350
13	17.06	25	24	600	0.009	0.011	0.079	-0.018	0.018	0.009	0.159	0.079	400
14	17.06	25	24	600	0.009	0.009	0.079	0.000	0.013	0.008	0.115	0.044	400
15	17.08	27	24	600	0.010	0.011	0.088	-0.009	0.015	0.012	0.132	0.026	400
16	17.08	27	24	600	0.012	0.012	0.106	0.000	0.017	0.013	0.150	0.035	400

Test T73 Rotary Motion Test at 600 Bar at Ambient Temperatures for Increasing Number of Cycles.

ROTARY MOTION TEST. **DATE** **13.8.97**

CALIBRATION FACTOR= **19.600** **[1 VOLT =2Kg]** **TEST NUMBER T73**
LEVER ARM (m)= **0.450**

Test Information					Number 1 out				Number 4 in				
Stroke	**Time when measurements taken.**	**Elapsed time**	**Temp**	**Press. Bar**	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Number of Cycles
1	17.19	0	25	600	0.018	0.223	0.159	-1.808	0.390	0.018	3.440	3.281	600
2	17.19	0	25	600	0.018	0.284	0.159	-2.346	0.362	0.018	3.193	3.034	600
3	17.25	6	25	600	0.018	0.264	0.159	-2.170	0.293	0.018	2.584	2.426	600
4	17.25	6	25	600	0.019	0.275	0.168	-2.258	0.291	0.020	2.567	2.390	600
5	17.35	16	25	600	0.022	0.238	0.194	-1.905	0.434	0.025	3.828	3.607	800
6	17.35	16	25	600	0.022	0.313	0.194	-2.567	0.368	0.023	3.246	3.043	800
7	17.41	22	25	600	0.023	0.302	0.203	-2.461	0.338	0.023	2.981	2.778	800
8	17.41	22	25	600	0.023	0.309	0.203	-2.523	0.342	0.024	3.016	2.805	800
9							0.000	0.000			0.000	0.000	
10							0.000	0.000			0.000	0.000	
11							0.000	0.000			0.000	0.000	
12							0.000	0.000			0.000	0.000	
13							0.000	0.000			0.000	0.000	
14							0.000	0.000			0.000	0.000	
15							0.000	0.000			0.000	0.000	
16							0.000	0.000			0.000	0.000	

Test T8 Linear Motion Test at Ambient Temperature and Different Pressures. New Seals.

LINEAR MOTION TEST.

CALIBRATION FACTOR= 2000 [1 VOLT =2000N] TEST NUMBER T8 DATE 15.8.97

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N	Voltage	Zero Voltage	Force Excluding Zero N	Force Including Zero N
1	18.26	0	22	0	-0.096	0.065	-192	-322	-0.148	-0.080	-296	-136	0.156	-0.013	312	338	0.173	0.064	346	218
2	18.28	2	22	0	-0.135	0.065	-270	-400	-0.116	-0.047	-232	-138	0.166	-0.080	332	492	0.174	0.480	348	-612
3	18.30	4	23	100	-0.282	0.041	-564	-646	-0.271	-0.150	-542	-242	0.314	-0.153	628	934	0.328	0.141	656	374
4	18.31	5	23	100	-0.252	0.132	-504	-768	-0.245	-0.140	-490	-210	0.324	-0.139	648	926	0.320	0.131	640	378
5	18.34	8	23	200	-0.347	0.068	-694	-830	-0.346	-0.168	-692	-356	0.390	-0.188	780	1156	0.416	0.172	832	488
6	18.34	8	23	200	-0.328	0.149	-656	-954	-0.327	-0.160	-654	-334	0.417	-0.145	834	1124	0.405	0.155	810	500
7	18.36	10	23	300	-0.410	0.062	-820	-944	-0.425	-0.185	-850	-480	0.413	-0.194	826	1214	0.455	0.173	910	564
8	18.37	11	23	300	-0.395	0.152	-790	-1094	-0.407	-0.170	-814	-474	0.467	-0.166	934	1266	0.461	0.164	922	594
9	18.39	13	23	350	-0.419	0.074	-838	-986	-0.445	-0.193	-890	-504	0.495	-0.168	990	1326	0.468	0.150	936	636
10	18.40	14	25	350	-0.420	0.143	-840	-1126	-0.443	-0.194	-886	-498	0.502	-0.179	1004	1362	0.504	0.159	1008	690
11	18.41	15	25	400	-0.486	0.074	-972	-1120	-0.434	-0.198	-868	-472	0.492	-0.180	984	1344	0.515	0.164	1030	702
12	18.43	17	25	400	-0.442	0.157	-884	-1198	-0.482	-0.165	-964	-634	0.549	-0.180	1098	1458	0.498	0.160	996	676
13	18.45	19	25	500	-0.601	0.069	-1202	-1340	-0.582	-0.235	-1164	-694	0.607	-0.240	1214	1694	0.526	0.141	1052	770
14	18.46	20	25	500	-0.580	0.163	-1160	-1486	-0.574	-0.220	-1148	-708	0.615	-0.225	1230	1680	0.563	0.173	1126	780
15	18.47	21	25	600	-0.673	0.070	-1346	-1486	-0.611	-0.249	-1222	-724	0.671	-0.220	1342	1782	0.592	0.158	1184	868
16	18.48	22	25	600	-0.631	0.144	-1262	-1550	-0.625	-0.247	-1250	-756	0.625	-0.220	1250	1690	0.580	0.184	1160	792

Test T9 Rotary Motion Test at Ambient Temperature and Different Pressures. New Seals.

ROTARY MOTION TEST. DATE 15.8.97

CALIBRATION FACTOR= 19.600 [1 VOLT =2Kg]
LEVER ARM (m)= 0.450

TEST NUMBER T9

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm
1	18.55	0	25	0	0.013	0.013	0.115	0.000	0.011	0.014	0.097	-0.026	0.021	0.010	0.185	0.097	0.030	0.014	0.265	0.141
2	18.56	1	25	0	0.015	0.015	0.132	0.000	0.012	0.015	0.106	-0.026	0.020	0.013	0.176	0.062	0.026	0.015	0.229	0.097
3	18.58	3	25	100	0.015	0.015	0.132	0.000	0.013	0.015	0.115	-0.018	0.021	0.013	0.185	0.071	0.022	0.018	0.194	0.035
4	18.59	4	25	100	0.016	0.017	0.141	-0.009	0.016	0.017	0.141	-0.009	0.021	0.017	0.185	0.035	0.029	0.017	0.256	0.106
5	19.00	5	25	200	0.016	0.017	0.434	0.025	0.015	0.017	0.132	-0.018	0.018	0.014	0.159	0.035	0.023	0.017	0.203	0.053
6	19.01	6	25	200	0.017	0.018	0.150	-0.009	0.014	0.017	0.123	-0.026	0.019	0.014	0.168	0.044	0.019	0.015	0.168	0.035
7	19.03	8	25	300	0.016	0.017	0.141	-0.009	0.013	0.016	0.115	-0.026	0.017	0.012	0.150	0.044	0.022	0.017	0.194	0.044
8	19.04	9	25	300	0.017	0.017	0.150	0.000	0.016	0.016	0.115	-0.026	0.019	0.017	0.168	0.018	0.019	0.017	0.168	0.018
9	19.06	11	25	350	0.019	0.019	0.168	0.000	0.017	0.018	0.141	-0.018	0.020	0.016	0.176	0.035	0.018	0.016	0.159	0.018
10	19.07	12	25	350	0.014	0.017	0.000	-0.026	0.006	0.011	0.150	0.053	0.026	0.011	0.159	0.062	0.023	0.018	0.203	0.044
11	19.09	14	25	400	0.016	0.017	0.141	-0.009	0.015	0.016	0.053	-0.088	0.018	0.016	0.221	0.079	0.023	0.016	0.203	0.062
12	19.10	15	25	400	0.015	0.016	0.132	-0.009	0.016	0.016	0.132	-0.009	0.025	0.017	0.168	0.018	0.023	0.018	0.203	0.044
13	19.12	17	25	500	0.017	0.017	0.150	0.000	0.012	0.017	0.141	-0.009	0.019	0.013	0.176	0.062	0.024	0.017	0.212	0.062
14	19.13	18	25	500	0.015	0.017	0.132	-0.018	0.016	0.016	0.106	-0.035	0.020	0.018	0.185	0.026	0.023	0.016	0.203	0.062
15	19.15	20	25	600	0.018	0.018	0.159	0.000	0.016	0.018	0.141	-0.018	0.021	0.017	0.176	0.026	0.027	0.019	0.238	0.071
16	19.16	21	25	600	0.018	0.019	0.000	0.000	0.017	0.018	0.150	-0.009	0.020	0.017	0.176	0.026	0.025	0.017	0.221	0.071

Test T10 Rotary Motion Test at Ambient Temperature and Different Pressures for a Lighter Lever Arm.

ROTARY MOTION TEST. **DATE** **18.8.97**

CALIBRATION FACTOR= **19.600 [1 VOLT =2Kg]** **TEST NUMBER T10**
LEVER ARM (m)= **0.450**

Test Information					Number 1 out				Number 2 out				Number 3 In				Number 4 In			
Stroke	Time when measurement taken.	Elapsed time	Temp	Press. Bar	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm	Voltage	Zero Voltage	Torque Excluding Zero Nm	Torque Including Zero Nm
1	14.14	0	22	0	0.417	0.487	3.678	-0.617	0.407	0.418	3.590	-0.097	0.441	0.406	3.890	0.309	0.503	0.441	4.436	0.547
2	14.17	3	23	0	0.418	0.503	3.687	-0.750	0.406	0.419	3.581	-0.115	0.429	0.405	3.784	0.212	0.503	0.427	4.436	0.670
3	14.45	0	26	100	0.423	0.501	3.731	-0.688	0.418	0.423	3.687	-0.044	0.422	0.419	3.722	0.026	0.510	0.421	4.498	0.785
4	14.47	2	26	100	0.420	0.506	3.704	-0.759	0.418	0.420	3.687	-0.018	0.423	0.419	3.731	0.035	0.511	0.422	4.507	0.785
5	14.49	4	26	200	0.419	0.510	0.434	0.025	0.418	0.419	3.687	-0.009	0.423	0.418	3.731	0.044	0.508	0.423	4.481	0.750
6	14.50	5	26	200	0.418	0.508	3.687	-0.794	0.417	0.419	3.678	-0.018	0.420	0.417	3.704	0.026	0.462	0.420	4.075	0.370
7	14.52	7	26	300	0.417	0.461	3.678	-0.388	0.416	0.417	3.669	-0.009	0.421	0.416	3.713	0.044	0.477	0.419	4.207	0.512
8	14.54	9	26	300	0.417	0.475	3.678	-0.512	0.416	0.417	3.669	-0.009	0.420	0.416	3.704	0.035	0.493	0.420	4.348	0.644
9	14.55	10	26	350	0.418	0.493	3.687	-0.662	0.417	0.418	3.669	-0.018	0.421	0.417	3.713	0.035	0.507	0.420	4.472	0.767
10	14.56	11	26	350	0.426	0.506	0.000	-0.706	0.417	0.426	3.678	-0.079	0.420	0.417	3.713	0.035	0.507	0.420	4.472	0.767
11	14.58	13	26	400	0.422	0.506	3.722	-0.741	0.417	0.422	3.678	-0.044	0.421	0.418	3.722	0.035	0.494	0.420	4.357	0.653
12	14.59	14	26	400	0.418	0.494	3.687	-0.670	0.418	0.418	3.678	-0.009	0.422	0.418	3.713	0.026	0.505	0.420	4.454	0.750
13	15.00	15	26	500	0.422	0.505	3.722	-0.732	0.418	0.423	3.687	-0.044	0.421	0.418	3.713	0.026	0.505	0.421	4.454	0.741
14	15.01	16	26	500	0.421	0.505	3.713	-0.741	0.418	0.421	3.687	-0.026	0.421	0.418	3.713	0.026	0.504	0.421	4.445	0.732
15	15.03	18	26	600	0.419	0.504	3.696	-0.750	0.418	0.419	3.687	-0.009	0.421	0.418	3.722	0.035	0.469	0.421	4.137	0.423
16	15.04	19	26	600	0.419	0.468	0.000	0.000	0.418	0.420	3.687	-0.018	0.422	0.418	3.722	0.035	0.505	0.421	4.454	0.741

APPENDIX F BASIC THICK WALL PRESSURE VESSEL STRESSING

BS 5500 (Ref 107) applies to thin cylinders where $\frac{OD}{ID} < 1.3$

Roark (Ref 108) defines a thin cylinder as $\frac{ID}{e} < 10$

Where:

OD = outer diameter
ID = inner diameter
t = wall thickness

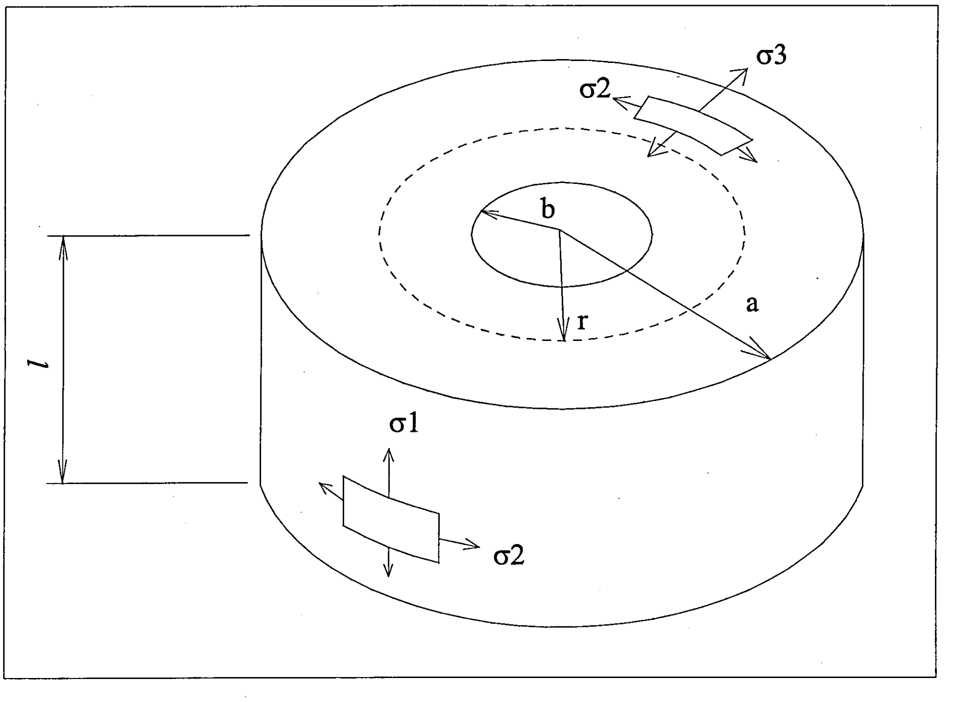


FIGURE F1 THE PRINCIPAL STRESSES IN A THICK WALLED PRESSURE VESSEL

where:
a = outer diameter
b = inner diameter
l = length of cylinder
r = radial position in cylinder wall
σ = principal stress in directions 1, 2 or 3.

Roark (Ref 108) quotes the following principal stresses for the thick walled cylinder, shown in Figure F1, under uniform internal pressure, q, in all directions; ends capped:

$$\sigma_1 = \frac{qb^2}{(a^2 - b^2)}$$

$$\sigma_2 = \frac{qb^2\,(a^2 + r^2)}{r^2\,(a^2 - b^2)}$$

$$\sigma_3 = \frac{-qb^2(a^2 - r^2)}{r^2(a^2 - b^2)}$$

The maximum stress occurs at $r = b$.

$$\therefore \sigma_{1max} = \frac{qb^2}{(a^2 - b^2)} \qquad \sigma_{2max} = q\frac{(a^2 + b^2)}{(a^2 - b^2)} \qquad \sigma_{3max} = -q$$

Applying Von Mises failure criteria:

$$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 = 2\sigma_{eff}^2$$

then to prevent failure : $\sigma_{eff} < f$, the material design stress.

Therefore, substituting σ_{1max}, σ_{2max} and σ_{3max} into Von Mises equation:

$$\left(q\frac{b^2}{a^2-b^2} - q\frac{(a^2+b^2)}{(a^2-b^2)}\right)^2 + \left[q\frac{(a^2+b^2)}{(a^2-b^2)} - (-q)\right]^2 + \left[-q - \frac{q\,b^2}{(a^2-b^2)}\right]^2 = 2\sigma_{eff}^2$$

$$\frac{q^2}{(a^2-b^2)^2}\left[a^4 + (2a^2)^2 + (-a^2)^2\right] = 2\sigma_{eff}^2$$

$$\frac{q^2}{(a^2-b^2)^2}\,6\,a^4 = 2\sigma_{eff}^2$$

Therefore the design stress $f > \sqrt{3}\,q\,\dfrac{a^2}{(a^2-b^2)}$ for a thick walled cylinder.

APPENDIX G DETAILED MANIPULATOR DESIGN

G1 INTRODUCTION

This Appendix presents the main manipulator design calculations and should be read in conjunction with Chapters 7 and 8. The calculations aim to demonstrate that the basic manipulator concept can be developed into a working tool.

Detailed drawings for the most important components are also presented to illustrate the calculations. They are not manufacturing drawings, but, with the text, contain sufficient detail to enable manufacturing drawings to be produced.

Note that unlike the component diameters, the length dimensions are generally not critical. Some of the length dimensions shown are therefore not round numbers, but they give a good impression of overall size. They can be varied if required.

G1.1 Initial Sizing

The OD of the manipulator must be less than 2.18" (55 mm). Assume an OD of 54 mm. For $f = 250$ MN/m^2 and $P = 744$ bar, see Section 8.3.1, then using Equation 8.1:

$$250\times10^6 > \sqrt{3}\times744\times10^5\times\frac{(54)^2}{[(54)^2 - (ID)^2]}$$

$$ID^2 < 1413$$

$$\underline{ID < 37.6 \text{ mm}}$$

The ID of the outer cylinder, away from the seals and thread connection must be less than 37.6 mm.

Access to machine the seal housings is difficult, as shown in Figure 7.11.2. A separate body is therefore required to house them.

To incorporate the main seals, as shown in Figure 7.9.1, at least 8.2 mm wall is required. The outer point of the seal must be within the 37.6 mm diameter so that there is still sufficient wall to hold the pressure. The maximum OD of the inner cylinder, on which the main seals seal, is therefore:

$$= 37.6 - 2\,(8.2)$$
$$= \underline{21.2 \text{ mm}}$$

Assume the OD of the inner cylinder is 20 mm. There is a 0.15 mm clearance between the inner cylinder and outer cylinder. The pressure diameter of the outer cylinder is therefore

$$= 20.3 + 2\,(8.2)$$
$$= \underline{36.7 \text{ mm}}$$

The minimum OD of the outer cylinder is then 53 mm (using equation 8.1 again).

G1.2 Thread Selection for Outer Cylinder

The root diameter of the external thread must be greater than the outer diameter of the seal cartridge. That is, the root diameter must be greater than 40 mm. The closest standard metric thread is an M45 with 1.5 mm pitch. This has an approximate root diameter on the male part of 42.3 mm.

The load carried by the threads during normal operation

= pressure force over a diameter of 36.7 mm (assume 37 mm) + frictional force due to translating the inner cylinder

$$= \left[\frac{\pi}{4}\times(37\times10^{-3})^2\times744\times10^5\right] + [15000]$$

$$= 94995$$

$$\approx \underline{100 \text{ kN}}$$

The critical direct stress areas are:

<u>Female</u> $\quad A_f = \frac{\pi}{4}\left[(53\times10^{-3})^2 - (45\times10^{-3})^2\right] = 6\times10^{-4}\ m^2$

<u>Male</u> $\quad A_m = \frac{\pi}{4}\left[(43.3\times10^{-3})^2 - (20.3\times10^{-3})^2\right] = 1.15\times10^{-3}\ m^2$

∴ The female part is critical.

The maximum applied load $F \quad < \quad A_f f$

$$< 6 \times 10^{-4} \times 250 \times 10^6$$
$$< \underline{150{,}000 \text{ N}}$$

∴ the factor of safety (equivalent *to* k_t) $= \dfrac{150{,}000}{100{,}000}$

$$= \underline{1.5}$$

150 kN maximum applied load is 70% of the minimum load capacity of the coiled tubing. This is satisfactory. There is therefore no problem with failure of the threads for short term loading. Fatigue failure due to the stress concentrations could be a problem.

The threads must therefore be examined regularly. The only other alternative is to increase the material design stress to 400 MN/m^2.

Thread Stripping

Equation 8.4 defines the load required to strip the threads:

The maximum applied load

$$F < 0.435\,\pi\, d\, t\, f$$
$$150{,}000 < 0.435\,\pi\,(45 \times 10^{-3})\, t\,(250 \times 10^{6})$$
$$t > 0.098$$
$$t > \underline{10\ \text{mm}}$$

This length of thread is much shorter compared to ⅞ times the thread diameter (40 mm) quoted in Section 8.2.3. This is because the thread has been selected to fit the available space. For safety, the length of thread should be at least 20 mm. (The seal rig contains 20 mm long thread connections).

Finally, for this section of the outer cylinder, the maximum ID of the outer cylinder away from the threads should be calculated. Basically there should be sufficient wall thickness to carry a maximum direct load of 150,000 N, i.e.

$$150{,}000 > \frac{\pi}{4}\left[(53 \times 10^{-3})^2 - (ID)^2\right] \times 250 \times 10^6$$

$$\underline{ID < 45\ \text{mm}}$$

This is not a problem, as the maximum ID to carry the internal pressure is 37.6 mm.

Figure G1.2.1 shows the Outer Cylinder connection.

G1.3 Sizing the Hexagon to Tighten the Outer Cylinder of the Manipulator

The hexagon must be placed as close to the threads as practicable. The outer cylinder ID, away from the seals, is 34 mm. The OD required to hold the pressure is therefore:

$$f > \sqrt{3}\ P\ \frac{OD^2}{(OD^2 - ID^2)}$$

$$250\times10^6 > \sqrt{3}\times744\times10^5\times\frac{OD^2}{OD^2 - 34^2}$$

$$1.94 > \frac{OD^2}{OD^2 - 34^2}$$

$$OD > 48.8$$

$$\underline{OD > 49\ mm}$$

A hex must therefore be machined with 49 mm A/F.

Figure G1.3.1 shows that the edges of the hex will have to be removed.

49 mm A/F is not a standard size and a separate tool would have to be made to tighten it.

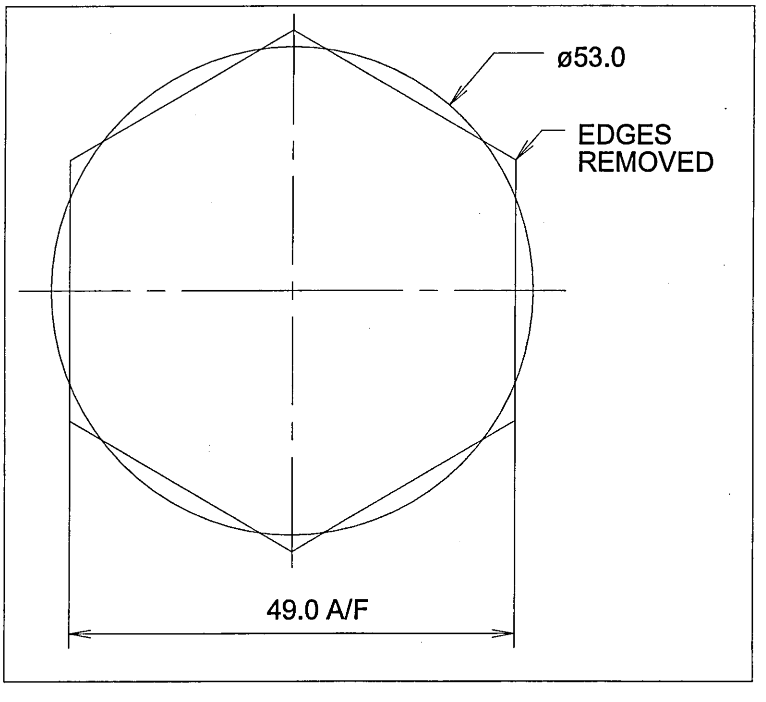


FIGURE G1.3.1 HEXAGON CONNECTION

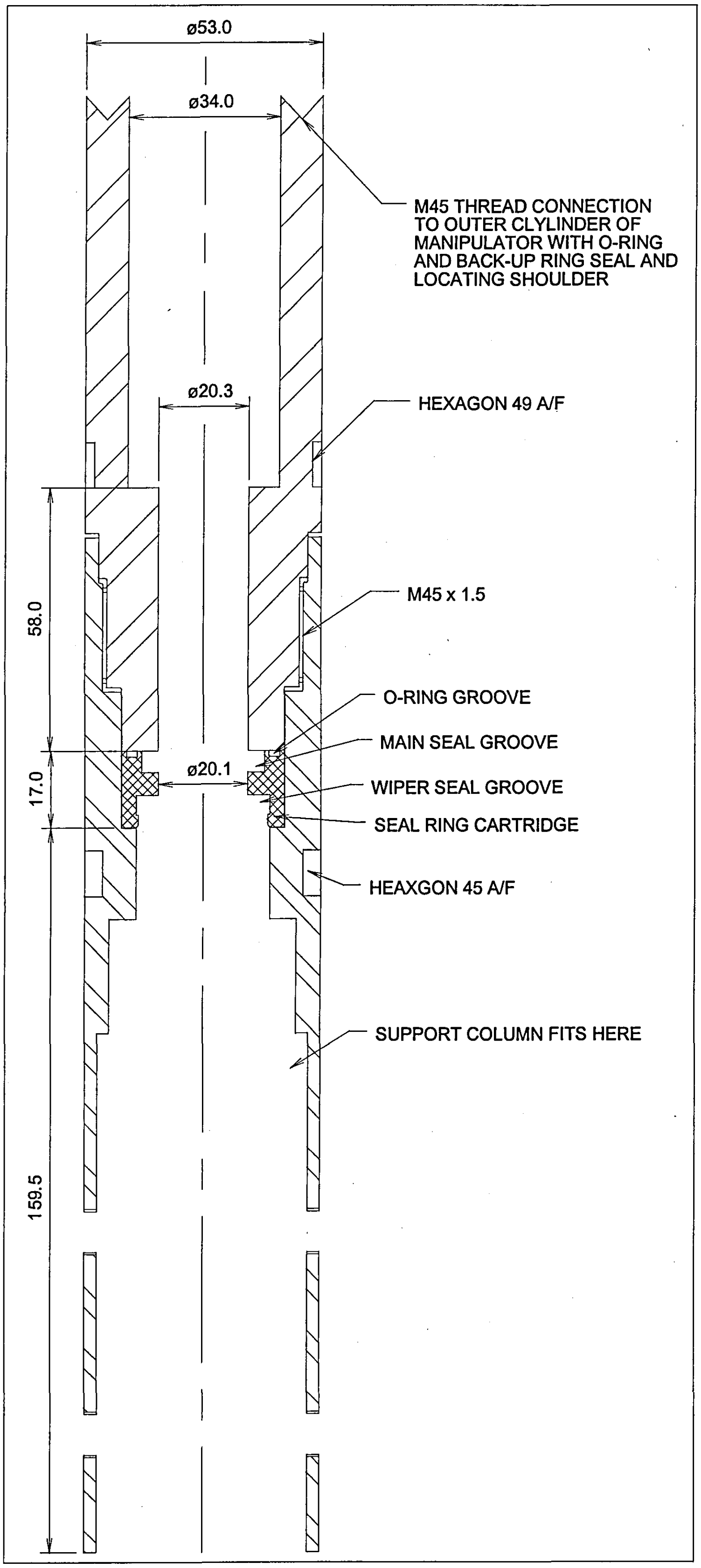


FIGURE G1.2.1 OUTER CYLINDER CONNECTION

G2 INNER CYLINDER DESIGN

G2.1 General Sizing

The OD of the inner cylinder is 20 mm. Using equation 8.1, the ID is given by:

$$250\times10^6 > \sqrt{3}\times744\times10^5\times\frac{(20)^2}{[(20)^2 - (ID)^2]}$$

$$\underline{\mathrm{ID} < 13.9\ \mathrm{mm}}$$

Figure 7.9.1 shows that the inner cylinder must be split in two places, either side of the nozzle holder, otherwise it cannot be assembled. There are several possible screw thread arrangements which incorporate an 'O' ring seal, but the resulting ID is in the order of 6 mm. If 100 l/min of abrasive cutting fluid is passing through the diameter, the abrasive velocity is 59 m/s. The inner cylinder will wear out rapidly.

The only alternative is to use the conventional method of sealing high pressure pipes and that is to seal on a 59° cone face (metal to metal seal).

The conventional high pressure connection would consist of a left hand thread on the pipe. A gland nut is passed onto the pipe and a collar screwed onto the left hand thread. The gland nut then screws into the housing, clamping the collar down and making the metal to metal cone seal.

Unfortunately, there is insufficient space to fit a gland in the nozzle housing. The threaded end of the inner cylinder must therefore screw directly into the nozzle holder. (This is the same as the nozzle holder connection used in the Cutting Head design, see Drawings CH02 and CH06 in Appendix D).

The inner cylinder will also be rotated. Grub screws are required to prevent the thread connection from unwinding. These were shown to work on the Seal Rig.

G2.2 Thread Selection

The minimum diameter that the inner cylinder has to pass is 20 mm, through the main seals. The threads must therefore be less than 20 mm and must be covered to protect the seals. The closest standard metric thread is an M18 and a 1.0 mm pitch gives a root diameter of 16.9 mm. Note that a fine pitch has been selected to give the maximum internal diameter.

The pressure force which is trying to separate the threads is over an area determined by the cone.

Assuming a bore diameter for the inner cylinder of 10 mm then the pressure force is:

$$= \frac{\pi}{4}\left[(16.9\times10^{-3})^2 - (10\times10^{-3})^2\right]\times744\times10^5 = 10846N \approx 11000N$$

The threads must also carry the translational frictional force = 15000 N.

The total maximum applied axial force = 26000 N

This load must be carried by the wall of the inner cylinder, A. Consequently the maximum axial load can be calculated:

Total axial force $\quad F \quad < \quad Af$

$$F < \frac{\pi}{4}\left[(16.9\times10^{-3})^2 - (10\times10^{-3})^2\right]\times 250\times10^6$$

$\underline{F < 36444 \text{ N}}$

This gives a safety factor of 1.4.

Under normal conditions, the working pressure would be much lower than 744 bar and consequently the safety factor is higher again.

G2.3 Thread Stripping

To prevent thread stripping,

$$\begin{aligned} F &< 0.435\,\pi\, d\, t\, f \\ 36444 &< 0.435\,\pi\,(18 \times 10^{-3})\, t\,(250 \times 10^6) \\ t &> 5.9 \times 10^{-3} \\ t &> \underline{6\text{ mm}} \end{aligned}$$

Again, this is much shorter than ⅞ d (16 mm). At least twice this thread length should be used.

G2.4 Selection of Grub Screws

Grub screws will be used to react the applied torque (they will also carry a contribution of the axial load).

The Fastener Handbook (Ref 114) quotes that a 3/16" (≈ 5 mm) BSW/BSF set screw with cup point has an axial holding power of 569 lbf (2530 N). A tightening torque of 32 lbf in (3.6 Nm) is required.

11.2 Nm was the predicted torque required to turn the inner cylinder (Section 7.10) The effective force applied at a radius of 10 mm is therefore 1120 N. One set screw would therefore be sufficient, although 2 or even 3 should be used to share the load. Access ports in the support column are required to allow the grub screws to be tightened.

G2.5 Nozzle Holder Sizing

The minimum diameter of the nozzle holder can be calculated by the fact that:

the direct stress area of the nozzle holder ≥ direct stress area of inner cylinder

$$\therefore \quad \frac{\pi}{4}\left(OD^2 - (18\times10^{-3})^2\right) \geq \frac{\pi}{4}\left[(16.9\times10^{-3})^2 - (10\times10^{-3})^2\right]$$

$$OD \geq 0.0226$$

$$\underline{OD \geq 22.6 \text{ mm}}$$

The maximum diameter of the nozzle holder depends on the maximum allowable ID of the support column.

The inner cylinder and nozzle holder connection is shown in Figure G3.2.

G2.6 Inner Cylinder Wear Protection

At least 100 l/min of abrasive fluid must pass through the 10 mm bore of the inner cylinder. The abrasive velocity is therefore 21 m/s. With an electric cable having to pass through the inner cylinder also, the abrasive velocity could be closer to 25 m/s.

Wear of the inner cylinder will be significant. As discussed in Section 7.14 a sacrificial liner should be installed in the inner cylinder and the electrical cable should be similarly protected. The liner should have a conical inlet to smoothly accelerate the flow, and can be held in place by a grub screw. This is shown in Figure G2.7.1.

G2.7 Incorporation of Bearing and Wiper in Inner Cylinder

Section 7.9 and Figure 7.9.1 show that a bearing and wiper seal need to be incorporated into the inner cylinder. These are shown in Figure G2.7.1. At the top of the inner cylinder is a hexagon (23 A/F, 25.4 A/C). There is sufficient space for a standard socket to fit over the hexagon to enable the inner cylinder to be tightened into the nozzle holder.

The inner cylinder wear liner can fit over the hexagon and can be attached by means of a grub screw. Note that a circular recess is machined into the flange of the liner. A hexagon would be very difficult to machine.

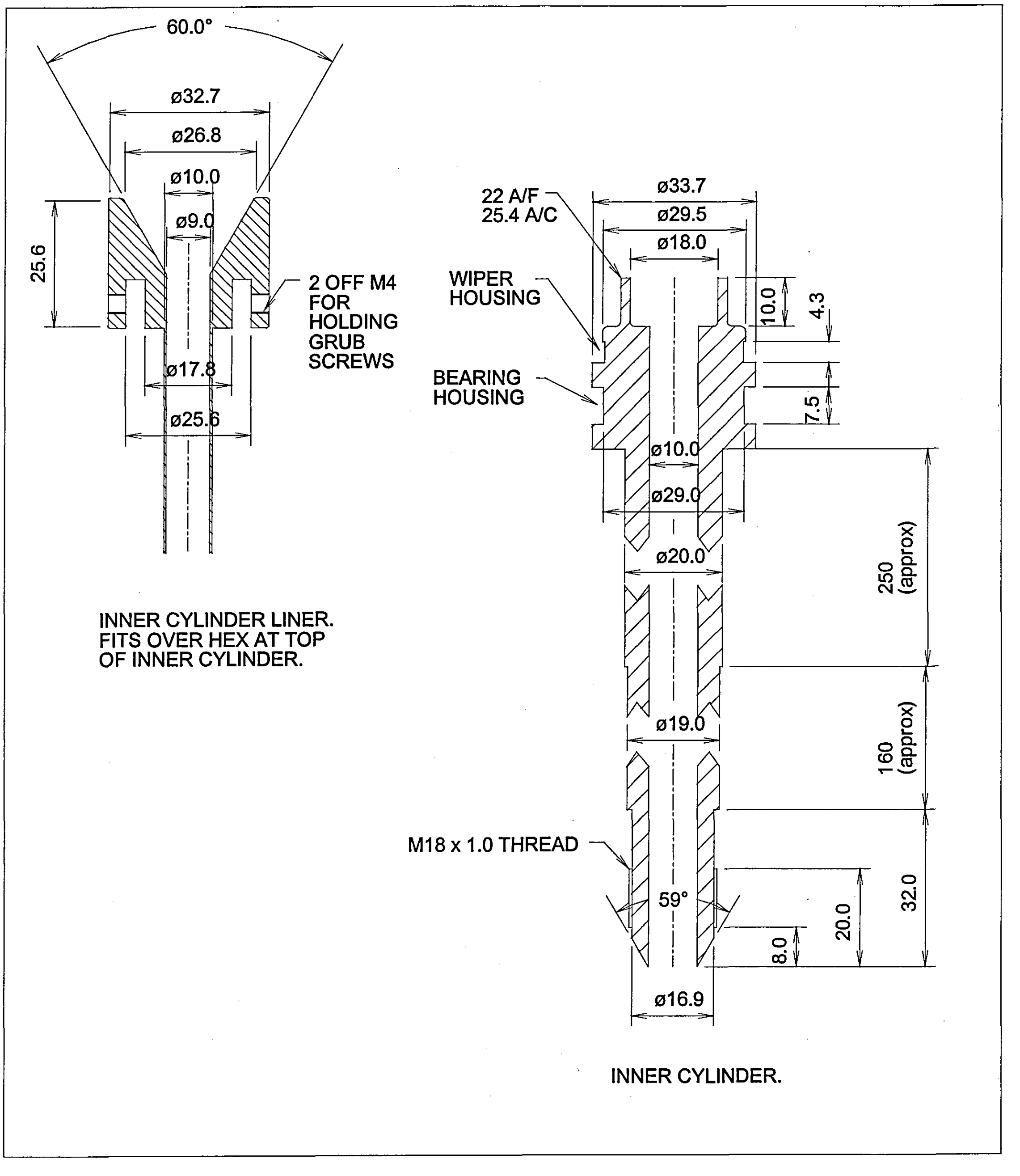


FIGURE G2.7.1 INNER CYLINDER AND LINER DESIGN

G2.8 Buckling of the Inner Cylinder

The critical buckling load for a simply supported column

$$P_{crit} = \frac{\pi^2 \, E \, I_{min}}{\ell_k^2}$$

where ℓ_k	=	length of inner cylinder, factored to account for the type of support
E	=	Young's Modulus
E	≈	200×10^9 N/m^2 for most steels
I_{min}	=	minimum polar moment of inertia

For the inner cylinder

$$I_{min} = \frac{\pi}{64}\,(OD^4 - ID^4)$$

$$= \frac{\pi}{64}\left((20\times10^{-3})^4 - (10\times10^{-3})^4\right)$$

$$= 7.36\times10^{-9}\ m^4$$

The maximum applied pressure load on the inner cylinder

$$F = \frac{\pi}{4}\left[(20\times10^{-3})^2 - (10\times10^{-3})^2\right](744\times10^5)$$

$$= 17530N$$

There is also a frictional load applied which was predicted to be $\frac{15000}{2}$ N

∴ Maximum applied load $= 25030$

$\approx$ <u>25000 N</u>

This is the critical buckling load.

Assuming the inner cylinder is simply supported at the seals, $\ell_k = \ell$ = length of the inner cylinder and:

$$\ell < \frac{\pi^2\times200\times10^9\times7.36\times10^{-9}}{25\times10^3}$$

$$\underline{\ell < 0.581}$$

The maximum distance between the main seals to prevent buckling of the inner cylinder is 580 mm.

G3 NOZZLE HOLDER DESIGN

A standard patented 3 mm diameter DIAJET nozzle is shown in Figure G3.1. This basic design applies to the full range of nozzle diameters.

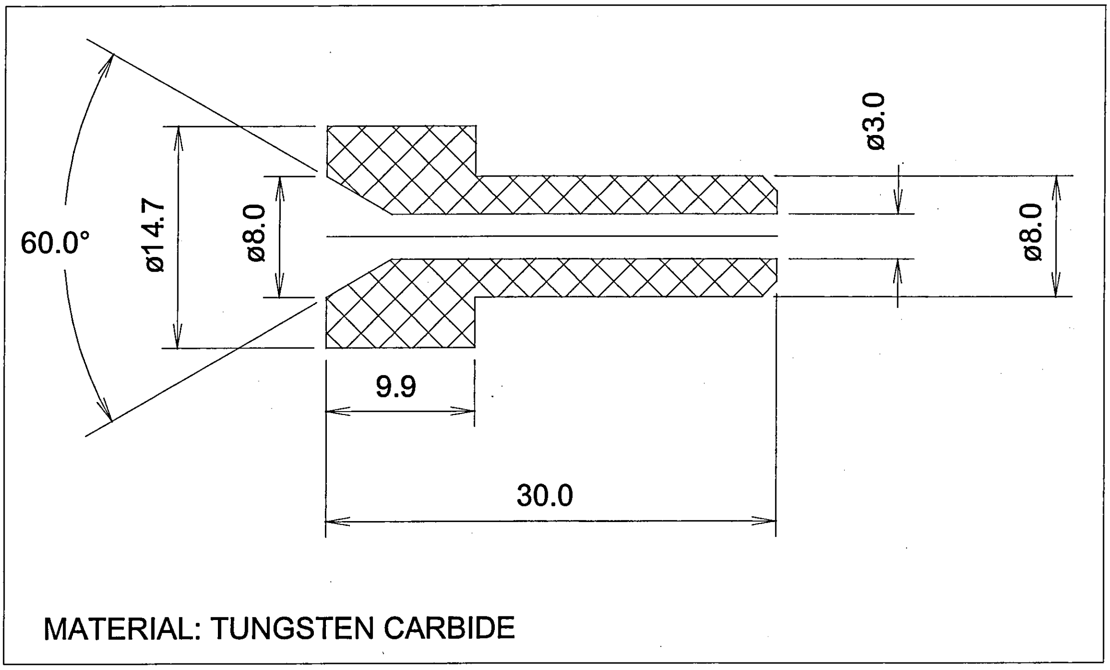


FIGURE G3.1 A STANDARD DIAJET NOZZLE

From the centreline of the manipulator to the OD is 26.5 mm. Therefore a 30 mm long nozzle will protrude past the centreline of the manipulator, blocking the bore of the nozzle holder. A shorter nozzle must be used.

There are several options for holding the nozzle in place.

1) Use a nozzle which is held in a screw-in fitting. This was used in the Cutting Head model.

2) Place the nozzle in a holder which has a screw on cap which fits over the nozzle.

The second option can be adapted so that the nozzle fits into a cavity in the nozzle holder and the cap then screws into the cavity, holding the nozzle firmly in place. Figure G3.2 shows this, the nozzle sitting in a recess of diameter 14.8 mm and the nozzle cap carrying an M16 x 1.0 (thread root diameter 14.99 mm).

Sealing the nozzle has been found to be not necessary. Even if water can escape around the back of the nozzle, once abrasive is passed through it, leakage stops, the abrasive blocking these escape pores. If leakage is a problem, an 'O' ring can be fitted behind the nozzle.

The maximum pressure force behind the nozzle

$$= \frac{\pi}{4}\,(15\times10^{-3})^2\times744\times10^5$$

$$= 13148\ N$$

The direct stress area on the nozzle cap

$$= \frac{\pi}{4}\left[(14.9\times10^{-3})^2\ -\ (8\times10^{-3})^2\right]$$

$$= 1.24\times10^{-4}\ m^2$$

The maximum pressure load it can therefore carry

$$= 1.24\times10^{-4}\times250\times10^6$$

$$= 31025\ N$$

The factor of safety $(k_t) = \dfrac{31025}{13148} = 2.36$ This is acceptable.

The nozzle holder must have at least the same direct stress area as the nozzle cap:

$$\frac{\pi}{4}\left[(OD)^2\ -\ (16\times10^{-3})^2\right] \geq 1.24\times10^{-4}$$

<u>OD nozzle holder ≥ 20.3 mm</u>

Note that in G2.5, the nozzle holder OD had to be greater than 22.6 mm.

If the maximum load that can be carried is 31025 N, the length of thread required is:

$$t > \frac{31025}{(0.435\ \pi(16\times10^{-3})\ (250\times10^{6}))}$$

$$\underline{t > 5.68mm}$$

Assume 6 mm.

The nozzle holder must have an OD of at least 22.6 mm. Assume a diameter of 25 mm. Figure G3.2 shows that the width of the holder from the centre to the edge of the nozzle is 18.5 mm. The holder can therefore be machined from rectangular bar. Flats are machined on the back of the holder.

The nozzle cap must carry an external hexagon, about 5 mm thick, to enable it to be tightened. The corners of the hexagon must not lie outside the 53 mm manipulator diameter. The most convenient standard size is a 17 mm hexagon across flats (maximum diameter 19.6 mm).

A wear plate must cover the nozzle cap. If the wear plate is made of a hardenable material, it can be recessed so that it fits over the nozzle cap. The wear plate must be of sufficient thickness to survive 20 hours of jet deflection, but must not lie outside the 53 mm manipulator diameter. The wear plate can be fixed onto the nozzle holder by grub screws. For additional protection, the nozzle cap could be hardened.

If an OD of greater than 53 mm is acceptable, then a longer nozzle can be used.

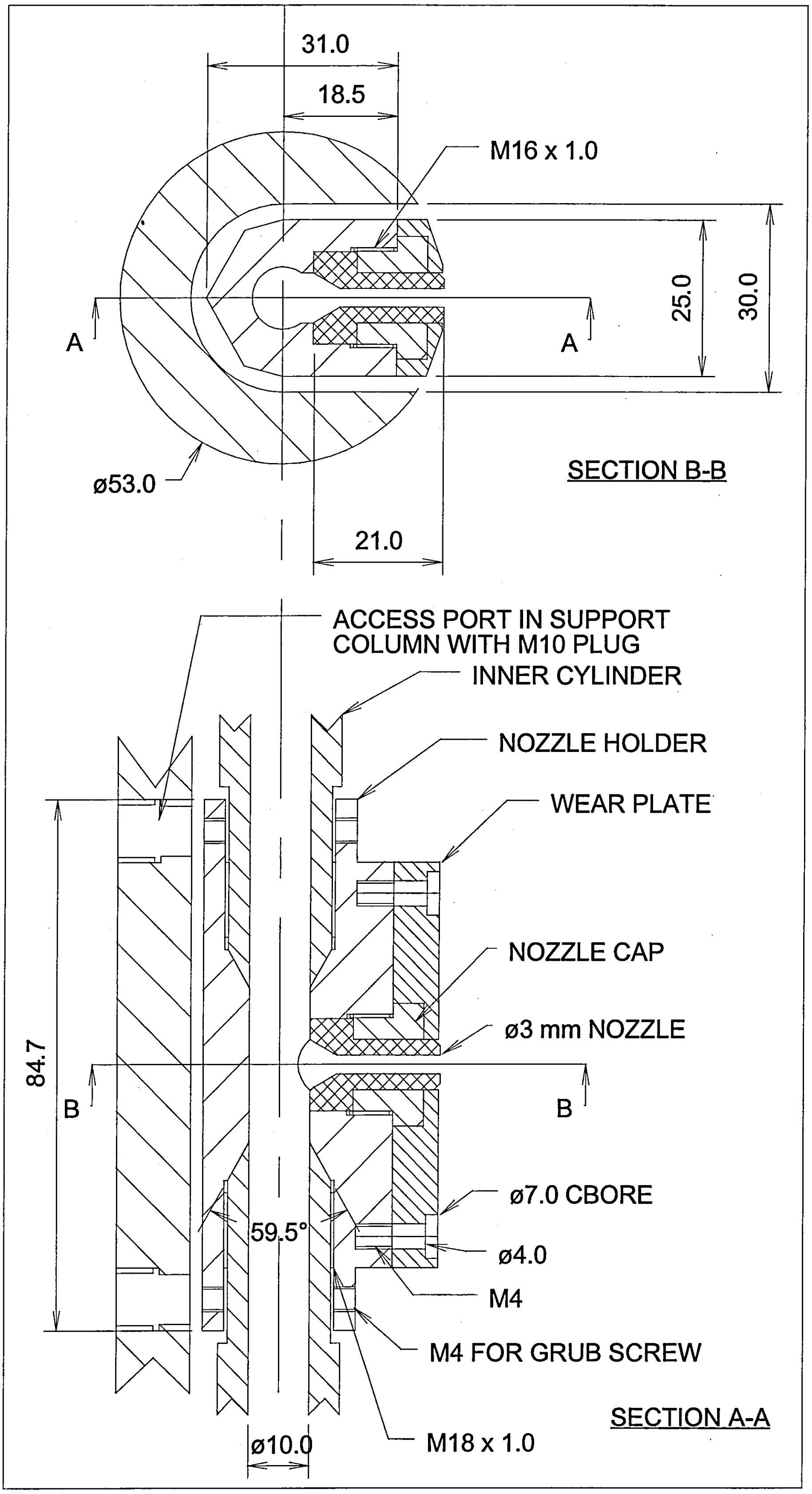


FIGURE G3.2 INNER CYLINDER AND NOZZLE HOLDER CONNECTION

G4 SUPPORT COLUMN DESIGN

Section 7.3 introduces the idea of a support column between the upper and lower sections of the manipulator. Bearings at either end of the support column allow it to rotate through 360° and ensure that the upper and lower sections of the manipulator are aligned. However the support column will also be in tension, for example when the manipulator is removed from the well. A series of load pins is therefore required to transmit the tensile loads (Figure G4.3.1). The actual loads that could be applied to the support column are detailed below.

G4.1 Loads Applied to the Support Column

There are two possible design arrangements that will affect the loads applied to the support column:

1) If cutting is only required over the stroke of the inner cylinder, no shear pin assembly is needed and the packer will support all of the loads. The support column is then in compression, transmitting the pressure and friction loads from the upper part of the manipulator to the packer. These loads can be reduced by reeling in the coiled tubing.

   The maximum compressive load = pressure force over the outer cylinder annulus + friction force due to the upper seals

   $$= \frac{\pi}{4}\left[(36.7\times10^{-3})^2 - (20\times10^{-3})^2\right](744\times10^5) + \frac{(15000)}{2}$$

   $$= 62{,}830 \text{ N}$$

   $$\approx \underline{63{,}000 \text{ N}}$$

   Note that the inner cylinder is pressure balanced and does not contribute to the support column load.

2) If the shear pin assembly is required, then the pressure force must be reacted by holding the coiled tubing and manipulator in tension.

   The support column must therefore react the pressure and friction forces associated with the lower part of the manipulator.

The maximum tensile load in the support column	=	Downwards pressure force over the bottom section of the outer cylinder + friction forces due to the lower seals

$$= \frac{\pi}{4}\,(10\times10^{-3})^2\,(744\times10^5) + \left(\frac{15000}{2}\right)$$

$$= 13343\ N$$

$$\underline{\approx 13500\ N}$$

Note that the lower part of the outer cylinder is pressure balanced, apart from the 10 mm diameter associated with the inner cylinder. The walls of the inner cylinder are also pressure balanced.

During cutting, the support column could therefore experience a maximum compressive load of 63,000 N and a maximum tensile load of 13500 N.

The support column must also transmit the high tensile load when the manipulator is removed from the well. In this case the column does not need to rotate and, if possible, the bearings should be decoupled so that they are not overloaded.

Section G1.2 shows that the maximum axial load that the manipulator body can carry is 150 kN. Ideally the support column, and therefore the loading pins holding the column in place, should also be able to carry this load.

G4.2 Bucking of the Support Column

The cross-sectional shape of the support column, away from the column ends, will be as shown in Figure G4.2.1.

To calculate the critical length of the support column under a compressive load of 63000 N, the position of the centre of gravity (COG) and moment of inertia (MI) about the centre of gravity need to be found.

The method of machining the slot in the support column must also be considered. The easiest method is to use a ball-nosed milling tool which will cut a radius at the bottom of the slot. The closest standard bull-nosed tool is 30 mm in diameter.

A clearance is required around the nozzle holder to allow assembly.

Roark (Ref 108) gives the relevant equations for properties of plane areas.

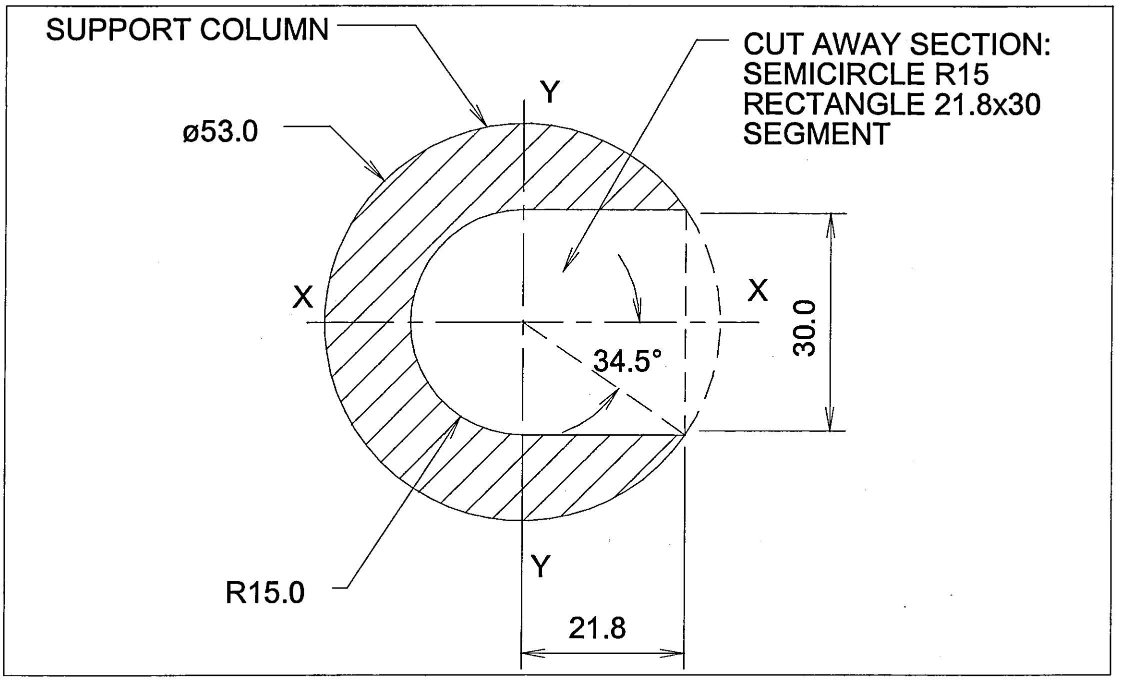


FIGURE G4.2.1 THE CROSS SECTIONAL SHAPE OF THE SUPPORT COLUMN AWAY FROM THE BEARING ASSEMBLIES

MI of Support Column About the Centreline Y-Y

$$MI_{supportY\text{-}Y} = MI_{circleY\text{-}Y}\ \phi 53 - MI_{semicircleY\text{-}Y}\ R15 - MI_{rectangleY\text{-}Y}\ (21.8x30) - MI_{segmentY\text{-}Y}$$

Circle

$$MI_{circleY\text{-}Y}\ \phi 53 = \frac{\pi\ (OD)^4}{64}$$

$$= \frac{\pi\ (53 \times 10^{-3})^4}{64}$$

$$= 3.87 \times 10^{-7}\ m^4$$

Semicircle

The MI of a semicircle about its central axis parallel to Y-Y is:

$$MI_{semicircle}\ R15 = 0.1098\,R^4$$

$$= 0.1098\,(15\times10^{-3})^4$$

$$= 5.6 \times 10^{-9}\ m^4$$

The distance of the central axis of the semicircle from Y-Y is:

$$Z_{semicircle} = 0.4244\,R$$

$$= 0.4244\,(15\times10^{-3})$$

$$= 6.37\times10^{-3}\ m$$

The area of the semicircle is:

$$A_{semicircle} = \frac{\pi}{2}\ (15\times10^{-3})^2$$

$$= 3.53\times10^{-4}\ m^2$$

The MI of the semicircle about Y-Y is:

$$MI_{semicircleY\text{-}Y} = MI_{semicircle} + A_{semicircle} Z^2_{semicircle}$$

$$= 5.6\times10^{-9} + (3.53\times10^{-4})(6.37\times10^{-3})^2$$

$$= 1.99\times10^{-8}\ m^4$$

Rectangle

The MI of a rectangle about its central axis parallel to Y-Y is:

$$MI_{rectangle} = \frac{b^3\ h}{12}$$

$$= \frac{(21.8\times10^{-3})^3(30\times10^{-3})}{12}$$

$$= 2.6\times10^{-8}\ m^4$$

The distance of the central axis of the rectangle from Y-Y is:

$$Z_{rectangle} = \frac{(21.8\times10^{-3})}{2}$$

$$= 10.9\times10^{-3}\ m$$

The area of the rectangle is:

$$A_{rectangle} = (21.8\times10^{-3})(30\times10^{-3})$$

$$= 6.54\times10^{-4}\ m^2$$

The MI of the rectangle about Y-Y is:

$$MI_{rectangleY\text{-}Y} = MI_{rectangle} + A_{rectangle}Z^2_{rectangle}$$

$$= 2.6\times10^{-8} + (6.54\times10^{-4})(10.9\times10^{-3})^2$$

$$= 1.04\times10^{-7}\ m^4$$

Segment

The MI of a segment about its central axis parallel to Y-Y is:

$$MI\ segment = (0.01143\ R^4\ \alpha^7)\ (1 - 0.3491(\alpha)^2 + 0.0450(\alpha)^4)$$

$$= (0.01143)\ (26.5\times10^{-3})^4\ (0.602)^7\ (1 - 0.3491(0.602)^2 + 0.0450(0.602)^4)$$

$$= (1.609\times10^{-10})(0.879)$$

$$= 1.4\times10^{-10}\ m^4$$

The distance of the central axis of the segment from Y-Y is:

$$Z_{segment} = (21.9\times10^{-3}) + 0.2R\alpha^2(1 - 0.0619\alpha^2 + 0.0027\alpha^4)$$

$$= (21.9\times10^{-3}) + 0.2(26.5\times10^{-3})(0.602)^2(1 - 0.0619(0.602)^2 + 0.0027(0.602)^4)$$

$$= 21.9\times10^{-3} + (1.92\times10^{-3})(0.98)$$

$$= 23.8\times10^{-3}\ m$$

The area of the segment is:

$$A_{segment} = \frac{2}{3}\ R^2\ \alpha^3\ (1 - 0.2\alpha^2 + 0.019\alpha^4)$$

$$= \frac{2}{3}\ (26.5\times10^{-3})^2\ (0.602)^3\ (1 - 0.2(0.602)^2 + 0.019(0.602)^4)$$

$$= 9.46\times10^{-5}\ m^2$$

The MI of the segment about Y-Y is:

$$MI_{segmentY\text{-}Y} = MI_{segment} + A_{segment}Z^2_{segment}$$

$$= 1.4\times10^{-10} + (9.46\times10^{-5})(23.48\times10^{-3})^2$$

$$= 1.4\times10^{-9}\ m^4$$

Support Column

$$MI_{supportY\text{-}Y} = MI_{circleY\text{-}Y} - MI_{semicircleY\text{-}Y} - MI_{rectangleY\text{-}Y} - MI_{segmentY\text{-}Y}$$

$$= 3.87\times10^{-7} - 1.99\times10^{-8} - 1.04\times10^{-7} - 1.4\times10^{-9}$$

$$= 2.6\times10^{-7}\ m^4$$

Position of the COG of the Support

Consider the section cut out of the support column.

The COG of the cut away section is given by:

$$A_{cutaway}\ Z = A_{semicircle}\ Z_{semicircle} + A_{rectangle}\ Z_{rectangle} + A_{segment}\ Z_{segment}$$

where A is the section area

Z is the vector distance from the position of the COG of the section to the centreline Y-Y

$$A_{cutaway} = 3.53\times10^{-4} + 6.54\times10^{-4} + 9.5\times10^{-5}$$

$$= 1.1\times10^{-3}\ m^2$$

$$(1.1\times10^{-3})(Z_{cutaway}) = (3.53\times10^{-4})\ (-6.37\times10^{-3}) + (6.54\times10^{-4})(10.9\times10^{-3}) + (9.5\times10^{-5})(23.8\times10^{-3})$$

$$= -2.25\times10^{-6} + 7.12\times10^{-6} + 2.26\times10^{-6}$$

$$= 7.1\times10^{-6}$$

$$Z_{cutaway} = 6.5\times10^{-3}\ m$$

Now $A_{support} Z_{support} + A_{cutaway} Z_{cutaway} = 0$

as the complete support has a circular cross-section.

$$A_{support} = \frac{\pi}{4}(53\times10^{-3})^2 - 1.1\times10^{-3}$$

$$= 1.1\times10^{-3}\ m^2$$

$$(1.1\times10^{-3})Z_{support} = -(1.1\times10^{-3})(6.5\times10^{-3})$$

$$Z_{support} = -6.5\times10^{-3}\ m$$

The COG of the support is 6.5 mm to the left of the Y - Y centreline.

<u>Moment of Inertia about the COG</u>

$$I_{COG} = I_{centreline} - A_{support} Z^2_{support}$$

$$= 2.62\times10^{-7} - (1.1\times10^{-3})\ (6.5\times10^{-3})^2$$

$$= 2.1\times10^{-7}\ m^4$$

Buckling of the Support Column

The support column must not buckle under a compressive load of 63,000 N (0.06 x 10^6 N)

$$P_{crit} = \frac{\pi^2 E I_{min}}{\ell_k^2}$$

ℓ_k is the length of the column, factored to account for the type of support
$E \approx 200 \times 10^9$ N/m^2
$I_{min} = 2.1 \times 10^{-7}$ m^4

Therefore, for a range of values for ℓ_k, the critical buckling load can be calculated. This is shown in Table G4.1.

TABLE G4.1 THE CRITICAL BUCKLING LOAD FOR DIFFERENT LENGTHS OF SUPPORT COLUMN

ℓ_k m	0.1	0.2	0.3	0.4	0.5	0.6	0.7
P_{crit} N x 10^6	41	10	4.6	2.6	1.6	1.1	0.8

The maximum length of the support column, ℓ, is determined by the maximum length of the inner cylinder. This is 0.58 m. Therefore, even in the worst support case of one end of the support column being free and $\ell_k = 2\ell$, buckling will not be a problem.

G4.3 Support Bearing Selection

The bearing assembly must be able to transmit a high thrust load, which could be compressive or tensile, and radial loads in order to maintain the alignment of the manipulator.

Available space is again the major problem. The OD of the inner cylinder is 20 mm. The ID of the support column in the region of the bearings must be greater than this. Assume a 1 mm clearance. The ID of the support column is therefore 22 mm.

The ID of the bearings must be greater than 22 mm and the OD must be much less than 53 mm in order to have sufficient space for a bearing housing. The SKF catalogue (Ref 119) details a variety of possible bearings, but to fit the space available, a thrust bearing and a needle roller bearing are the most suitable. Table G4.3.1 gives the most important bearing details.

TABLE G4.3.1 BEARING DETAILS

Bearing Type	ID mm	OD mm	C N	C_o N
Thrust Bearing	25	42	15900	31500
	25	47	27600	55000
	30	47	16800	36000
Needle roller bearing (sealed with inner ring)	20	37	19400	22400
	25	42	21600	27500
	30	47	23300	32000

C - Dynamic load rating based on a rating life of 1,000,000 revolutions

C_o - Static load rating; applicable to low speeds, stationary and shock loads.

Note that the thrust bearing requires pre-loading.

Note that the static load rating for some of the thrust bearings is much smaller than the maximum compressive load of 63,000 N. However, the bearings are designed for long life and should be able to carry loads as high as this for the 20 hours running time. If necessary, they can then be replaced.

The proposed bearing assembly is shown in Figure G4.3.1.

Working from the top down it consist of:

a) A sealed needle roller bearing (ϕ25 x ϕ42) which is held in place on a 26.9 mm diameter shoulder on the support column by a spiral retaining ring (ϕ23.42 x ϕ27.28 Ref 120).

Tolerance on support column: $\phi 25 \begin{smallmatrix} +0.011 \\ +0.002 \end{smallmatrix}$ (k5)

Tolerance on housing: $\phi 42 \begin{smallmatrix} +0.050 \\ +0.025 \end{smallmatrix}$ (F7)

Note the larger clearance on the housing (outer cylinder) to allow some axial motion when required.

b) M28 x 1.0 nut and locking nut (thread root diameter 26.9 mm), which holds the thrust bearing assembly together.

The direct stress area of the support column in the region of this nut is

$$= \frac{\pi}{4}\times\left[(26.9\times10^{-3})^2 - (22\times10^{-3})^2\right] = 1.88\times10^{-4}\ m^2$$

Maximum applied axial load $1.88\times10^{-4}\times250\times10^{6} = 47047\text{N}$

Assuming $k_t = 2.8$, then the applied axial load < 16800 N.

This is acceptable. The nut will only be loaded in tension and the maximum tensile load that the thrust bearing has to operate at is 13500 N.

For the same direct stress area the OD of the M28 nut must be 32 mm.

To prevent thread stripping, the length of thread

$$t > \frac{47047}{0.435\pi\ (28\times10^{-3})\ (250\times10^{6})}$$

$$\underline{t > 4.9mm}$$

Assume 10 mm.

c) 2 support washers (ϕ33.5 x ϕ47) which sandwich a disc spring. The disc spring is modified (standard disc spring ϕ31 x ϕ63 machined to ϕ31, ϕ47, Ref 121) to give a deflection of 0.45 mm at 15000 N. (By using additional disc springs, this deflection can be increased, or the load to give the same deflection can be increased).

d) Thrust bearing (ϕ30 x ϕ47). The thrust bearing sits on a spiral retaining ring (ϕ28.5 x ϕ32.9) (Ref 120) and is pre-loaded by the M28 nut.

Tolerance on support column: $\phi30\ {}^{+0.000}_{-0.033}$ (h6)

Tolerance on housing - a clearance is required between the housing and thrust bearing; the thrust bearing must not be radially loaded ⇒ ϕ47.6 - ϕ47.5

e) Loading pins screw into the outer body of the manipulator and locate in a groove in a loading ring (ϕ33.5 x ϕ47) which floats freely around the support column. When the manipulator outer body is lifted up when the coiled tubing is reeled in, the loading pin moves the loading ring upwards, lifting the thrust bearing off the spiral retaining ring so that it is free to rotate.

f) A second thrust bearing sits on a similar arrangement of washers and disc springs (ϕ30.5 x ϕ46.5) described in (c), which sit on a shoulder on the support column (OD 47 mm). The thrust bearing is pre-loaded by a spiral retaining ring ((ϕ28.5 x ϕ32.9) and the disc spring.

When the manipulator outer body moves down, the loading pins move the loading ring down, pushing the thrust bearing off the spiral retaining ring so that it is free to rotate.

'O' rings are housed in both faces of the load ring to hold it in position under no load. When load is applied, one of the 'O' rings compresses and the load ring transmits the pin load to the thrust bearing.

g) A second set of loading pins screw into the outer body of the manipulator and locate in a groove in the support column. There is a clearance between these loading pins and the top of the groove in the support column so that the pins do not normally engage. If the tensile load exceeds 15000 N, for example when the manipulator is removed from the well, the top disc spring on the support column will compress by 0.45 mm and this second set of loading pins will then engage. These pins then carry any additional tensile load, protecting the bearings and M28 nut from overload. Once the loading pins engage, the support column cannot be rotated.

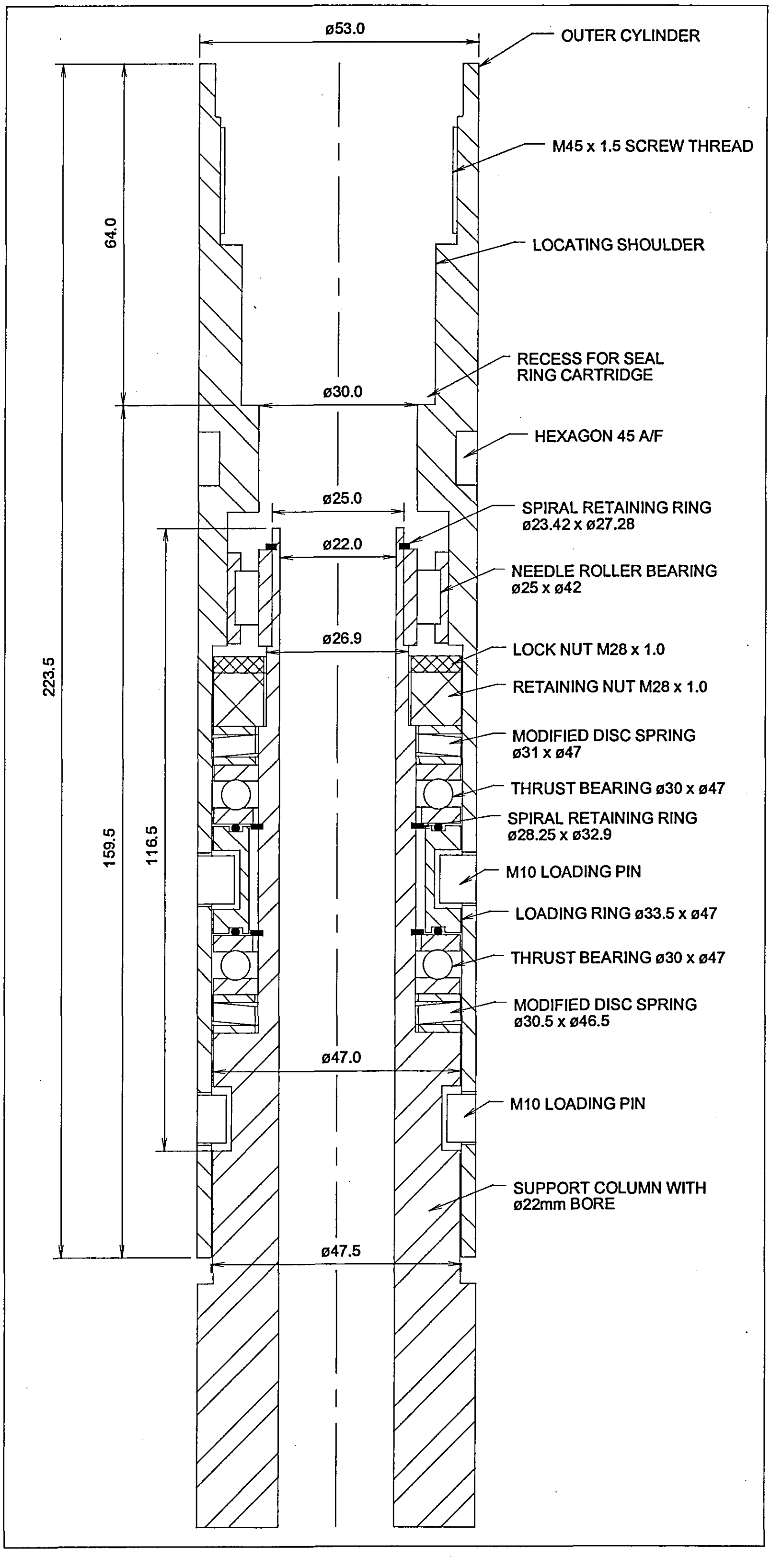


FIGURE G4.3.1 SUPPORT COLUMN BEARING ASSEMBLY

Spiral Retaining Ring Details:

Needle roller bearing retaining ring, which just holds the needle roller bearing in position:

Shaft diameter = 25 mm

Groove dimensions:

Diameter	23.42 mm ± 0.075
Depth	0.79 mm
Width	1.20 mm + 0.08, -0.00

Thrust capacity:	Ring shear = 27,990 N
	Groove yield = 9620 N

Thrust bearing retaining ring, which pre-loads the thrust bearings:

Shaft diameter = 30 mm

Groove dimensions:

Diameter	28.25 mm ± 0.01
Depth	0.88 mm
Width	1.40 mm +0.01, 0.00.

Thrust capacity:	Ring shear = 34,640 N
	Groove yield = 12860 N

G4.4 Loading Pin Design

The loading pins are critical to the guaranteed operation of the manipulator. There are two sets which engage the support column:

- the first set transmits the normal operating loads to the thrust bearing while cutting. The maximum load is 63,000 N in compression, although this load can be reduced by holding the coiled tubing in tension, and 15,000 N in tension..

- the second set transmit tensile loads exceeding the normal operating load of 15000 N. These pins must be rated to as high a load as possible.

Failure

The loading pin assembly can fail in 3 ways.

1) Pure tension

The critical area = Annulus area of the outer cylinder - Area lost due to the loading pin holes

$$= \frac{\pi}{4}\left[(53\times10^{-3})^2 - (47.5\times10^{-3})^2\right] - Ndt$$

$$= \underline{4.34 \times 10^{-4} - Ndt}$$

where N is the number of loading pins
d is the diameter of the pins
t is the wall thickness $= \left(\frac{53 - 47.5}{2}\right) = 2.75mm$

The design stress of the outer cylinder $\geq 250 \times 10^6$ N/m^2.

2) Pin Shear

The critical area = area of cross-section of the pin in shear.

BS 5950, Part 1, 1990 (Ref 122) gives the shear strength of a bolt $P_s = 0.48\ U_f$ but $\leq 0.69\ Y_f$

where U_f is the bolt tensile strength
Y_f is the bolt yield strength

Consider using a standard grade 12.9 bolt (Ref 114):

$$U_f = 120\ kg/mm^2 = 1.17\times10^9 N/m^2 \quad (0.48U_f = 562\times10^6 N/m^2)$$

$$Y_f\ is\ 90\%U_f = 1.05\times10^9 N/m^2 \quad (0.69Y_f = 724\times10^6 N/m^2)$$

Therefore the shear strength at room temperature = 562×10^6 N/m^2.

This strength will drop for operations at 200°C.

As a first assumption, reduce the shear strength by 30% to account for the high temperatures.

Then $P_s \approx \underline{393 \times 10^6 \text{ N/m}^2 \text{ at } 200°\text{C}}$.

3) Bearing

The critical area, which is the same for the bolt and the housing, = dt

d - diameter of the bolt
t - length of hole in housing

BS 5950, Part 1, 1990 (Ref 122) defines the bearing strengths for bolts and connected parts in clearance holes. For this application the holes are threaded.

Bearing strength for bolt $\quad P_{bb} = 0.72 \left(U_f + Y_f\right)$

For a grade 12.9 bolt $\quad P_{bb} = 0.72 \ (1.17\times10^9 + 1.05\times10^9)$

$$= 1.6 \times 10^9 \text{N/m}^2 \text{ at room temperature.}$$

Assuming a 30% reduction due to the high temperature

$$P_{bb} = 1.12\times10^9 \ \text{N/m}^2 \text{ at } 200°\text{C.}$$

Bearing strength for connected parts $P_{con} = 0.65\ (U_s + Y_s)$

Where U_s is the tensile strength of the part
Y_s is the yield strength of the part

If the design stress for the part is 250 x 10^6 N/m^2, Y_s is approximately 400 x 10^6 N/m^2 and U_s is 665 x 10^6 N/m^2 (Values taken for Ferralium at 200°C)

$$\therefore P_{con} = 0.65\ (400 + 665)$$
$$= \underline{692 \times 10^6 \text{ N/m}^2}$$

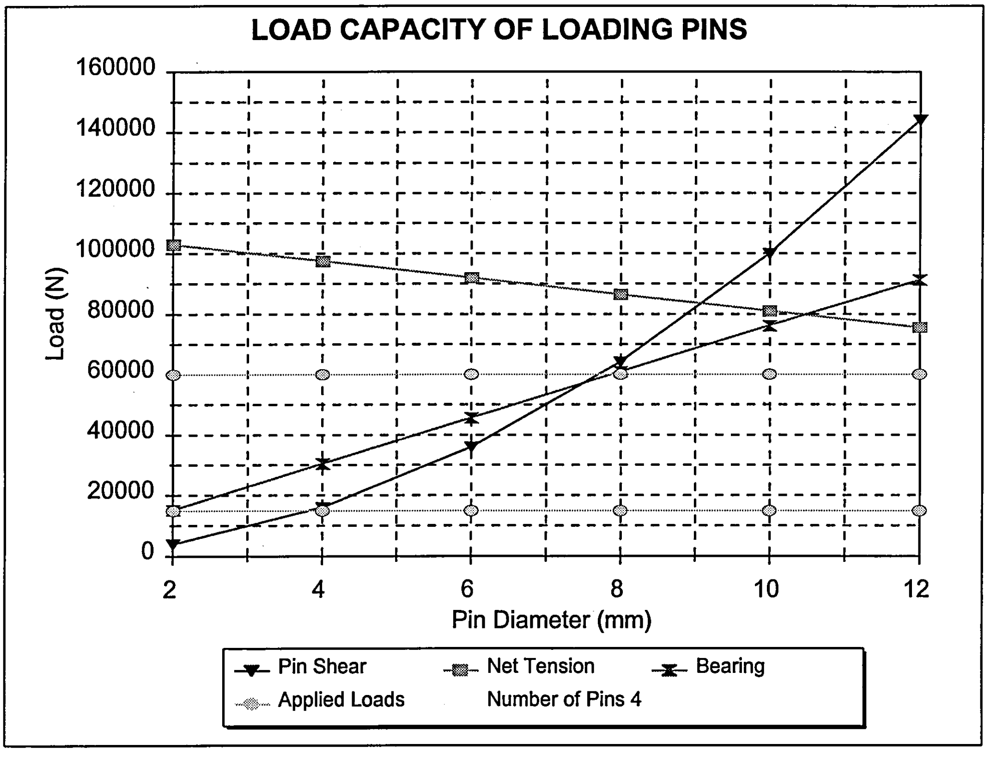
$P_{con} << P_{bb}$. The connected parts will failure in bearing first. Only this failure curve therefore needs to be considered.

Calculation of Failure Loads

The failure loads for tension, shear and bearing can be calculated on a spreadsheet using the respective allowable stresses for different bolt diameters and different numbers of bolts.

A set of curves of failure load against bolt diameter can then be plotted so that the optimum bolt diameter can be selected. Graph G4.4.1 shows the failure curves for 4 loading pins, which was found to be the optimum arrangement.

GRAPH G4.4.1



Analysis of these curves shows that:

- for the first set of loading pins, 4 x M10 bolts are suitable.
- it is not realistic to use even larger bolts for the second set of loading pins. 4 x M10 bolts should be used. These have a maximum load carrying capacity of just over 60,000 N. If the 15,000 N capacity of the first set of pins is included, then the maximum load available to pull the manipulator from the well is 75,000 N.

This pull load can be increased by changing the disc spring or adding two further disc springs so that the second set of loading pins do not engage until a load of 45,000 N. This is the maximum load that can be carried by the M28 nut which holds the thrust bearing in position.

The maximum pull load will then be 105,000 N, which is more reasonable.

Note that although the coiled tubing can pull up to 200,000 N, much of this force will be lost in friction and the manipulator will never experience such high loads. The top of the manipulator should be strain gauged so that the manipulator cannot be overloaded.

Loading Pin Pullout Through the Support Column

Let P = load per pin

If the 4 M10 pins can carry 60,000 N, the load per pin, P, is 15,000 N.

The area for pullout = $2ht$ (ht in the outer cylinder and the support column)

where t is the outer cylinder wall
h is the distance from the hole centreline to the end of the outer cylinder.

For no end shear out failure $0.58f > \dfrac{P}{2ht}$

f is the design stress $\geq 250 \times 10^6$ N/m^2, $0.58f$ is the design shear stress.

$$\therefore\ 0.58 \times 250 \times 10^6 \geq \frac{15000}{2h\left(\dfrac{(53\times10^{-3}) - (47.5\times10^{-3})}{2}\right)}$$

$$h \geq 0.019$$
$$\underline{h \geq 20\ mm}$$

For the loading ring,

$$t = \frac{(47\times10^{-3}) - (31\times10^{-3})}{2} = 8\times10^{-3}\ m$$

$$\therefore h \geq \frac{15000}{(0.58\times250\times10^{6}\times)(8\times10^{-3}\times2)}$$

$$\geq 6.5\times10^{-3}$$

$$\geq 6.5\ mm$$

The loading ring should therefore be twice this height, as the pin could shear out in both directions.

G4.5 Calculation of Bearing Friction

The SKF Catalogue (Ref 120) details equations for calculating the bearing friction. Of greatest importance is the starting torque. This is generally considered to be twice the load-dependent frictional moment M_1.

$$M_1 = f_1 \, P_1^a \, d_m^b$$

M_1	load dependent moment Nmm
f_1	function of bearing type and load
P_1	applied load N
d_m	mean diameter of bearing $\dfrac{(OD + ID)}{2}$ mm
a,b	function of bearing type

Additional frictional losses arise from rubbing seals. The frictional moment, M_3, can be higher than M_1.

$$M_3 = \left(\frac{OD + ID}{f_3} \right)^3 + f_4$$

f_3 and f_4 are factors.

<u>Needle Roller Bearing</u>

ID = 25 mm   OD = 42 mm

The radial load is unknown. Assume $P_1 = C$ the dynamic load rating = 21,600 N.

$f_1 = 0.002$
$a = 1$
$b = 1$
$d_m = 33.5$

$M_1 = 0.002 \text{ x } 21600 \text{ x } 33.5$
$= 1447$ Nmm
$\equiv$ <u>1.5 Nm</u>

$f_3 = 20$
$f_4 = 25$

$$M_3 = \left(\frac{25 + 42}{20}\right)^2 + 25$$

$$= 36 \text{ Nmm}$$

$$= \underline{0.036 \text{ Nm}}$$

$\therefore$ The load dependant torque for the roller bearing is 1.5 Nm.

<u>Thrust Bearing</u>

ID = 30 OD = 47

The maximum applied dynamic load = 15,000 N

$$f_1 = 0.0008\left(\frac{P_1}{C_o}\right)^{0.33} = 0.0008\left(\frac{15000}{36000}\right)^{0.33} = 6\times10^{-4}$$

$a = 1$
$b = 1$
$d_m = 38.5$

$M_1 = 6 \times 10^{-4} \times 15000 \times 38.5$
$= 347$ Nmm
$\equiv \underline{0.4 \text{ Nm}}$

$\therefore$ Total bearing friction for 2 thrust bearings and 2 needle bearings

$= 2\,(1.5 + 0.4)$
$= \underline{3.8 \text{ Nm}}$

The break-out torque could be as high as twice this value

$= \underline{7.6 \text{ Nm}}$

The results of the seal test rig show that a maximum break-out torque of 3.5 Nm was measured.

The minimum torque that the motor must generate is therefore 11 Nm.

Note that the break-out friction due to the needle roller bearings contributes considerably to this total torque. It is unlikely that the radial load on these bearings will be much higher than 3000 N (10% of the axial load) as only the bottom of the manipulator will be fixed in place.

For a radial load of 3000 N, $M_1 = 0.201$ Nm and the minimum break-out torque that the motor must generate is then 6 Nm.

<u>This needs to be simulated and measured.</u>

G5 THE STROKE OF THE INNER CYLINDER

Based on the dimension of the Support Column, Outer Cylinder Connection and Nozzle Holder, the nozzle stroke and the length of the upper and lower inner cylinders can be selected.

Section G2.8 showed that the maximum length of the inner cylinder between supports was 580 mm. Note that the supports are the bearings at the end of the inner cylinder and the main seals.

The inner cylinder connects to the nozzle holder by means of an M18 thread. The OD of the inner cylinder is 20 mm to fit the main seals. However, there is approximately 140 mm from the main seal to the top of the slot in the support column (Figure G4.3.1). Over this length, the inner cylinder will never meet the seals. The OD over the length should therefore be reduced very slightly, perhaps to 19 mm, so that the inner cylinder can be easily pushed through the main seals. (Note for 19 mm OD, the maximum ID to retain pressure is 13.2 mm).

There is a further distance of approximately 50 mm from the main seal to the flange in the bore of the outer cylinder, through which the end of the inner cylinder, carrying the wiper and bearing, cannot pass.

A 200 mm stroke is therefore possible. Each inner cylinder will then be over 400 mm long (Figure G2.7.1).

If the nozzle holder is 100 mm long, then the slot in the support column must be 300 mm long and the support column must be over 500 mm in length.

G5.1 Tightening the Inner Cylinder / Nozzle Holder Connection

The inner cylinder will be tightened into the nozzle holder by turning the top of the inner cylinder, on which a hex has been machined (Section G2.7), and holding the support column still. There is a worry that final tightening will over twist the inner cylinder, as the distance between the top and the nozzle holder is over 400 mm.

There is insufficient space between the inner cylinder and support to tighten the inner cylinder easily. Instead of machining a hexagon, a series of flats could be machined which are staggered over a short length of the inner cylinder.

The ID of the inner cylinder is 10 mm. The OD required to hold pressure is therefore:

$$f > \sqrt{3}\ P\ \frac{OD^2}{OD^2 - ID^2}$$

$$250\times10^6 > \sqrt{3}\times744\times10^5\times\frac{OD^2}{OD^2 - 10^2}$$

$$1.94 > \frac{D^2}{D^2 - 10^2}$$

$$OD > 14.4$$

$$\underline{\text{OD} > 15\ \text{mm}}$$

16 mm across flats can therefore be machined onto the inner cylinder. There is therefore a 7 mm clearance between the inner cylinder and the support column in which the tightening tool must fit and rotate. This needs to be tested to see if the flats are beneficial. Alternatively, a section of the support column will have to be removable to allow sufficient access to torque the connection properly.

G5.2 Bending Due to the Jet Reaction

The jet reaction force must be considered as this will reduce the critical buckling load of the inner cylinder. The jet reaction force, $R = 0.25\ Q\ \sqrt{P}$, where Q is the flow rate and P is the nozzle pressure drop in bar.

Typically we would expect $Q \approx 90$ l/min and $P \approx 300$ bar.

$$R = 0.25 \times 90 \times \sqrt{300}$$

$$R = \underline{390\ \text{N}}$$

The central deflection of a simply supported beam under a centrally acting load $\delta = \frac{R\ell^3}{48EI}$

ℓ	-	length of beam. In this case the maximum unsupported length is 0.58 m.
E	-	Young's Modulus of Elasticity ($\approx 200 \times 10^9$ for steel)
I	-	Moment of Inertia

For the inner cylinder:

$$I = \frac{\pi}{64}\left(OD^4 - ID^4\right)$$

$$I = \frac{\pi}{64}\left[(20\times10^{-3})^4 - (10\times10^{-3})^4\right]$$

$$\underline{I = 7.4 \text{ x } 10^{-9} \text{ m}^4}$$

Therefore:

$$\delta = \frac{390\times(0.58)^3}{64\ (7.4\times10^{-9})\ (200\times10^{9})}$$

$$\delta = 8\times10^{-4}\ m$$

$$\underline{\delta = 0.8 \text{ mm}}$$

This is a significant deflection. A low friction support should therefore be positioned behind the nozzle, which slides along the inside of the support column. This support will also increase the critical buckling load of the inner cylinder.

For example, if a PTFE support was used, the coefficient of friction, μ, between PTFE and steel is between 0.03 and 0.05 (Ref 123).

The frictional force to move the PTFE	=	μF
	=	0.05 x 390
	=	$\underline{19.5 \text{ N}}$

This is minimal compared to the seal frictional forces.

Abrasive bedding into the PTFE surface will increase this frictional force. The support will therefore need to be changed regularly (see Figure 7.17.1).

G6 LEAD SCREW SELECTION

A variety of companies manufacture suitable lead screws.

The critical criteria for selecting a lead screw are:

- the dynamic load rating C_a
- the static load rating C_{oa}
- the critical buckling load
- the frictional force to turn the lead screw
- the diameter of the lead screw nut, which must fit in the outer cylinder of the manipulator.

For the required load and friction characteristics, a planetary roller screw, such as the ones supplied by SKF (Ref 124), are the most suitable.

Note that planetary roller screws are reversible, they can be back-driven by an applied axial load. A brake is required on the motor to prevent this.

The inner cylinder and lead screw are pressure balanced. The only applied load is therefore due to the break-out seal friction forces. For these calculations, this has been estimated at 15000 N.

The critical buckling load

$$P_{crit} = \frac{\pi^2 \, E \, I_{\min}}{\ell_k^2}$$

where: E = Young's' Modulus for the lead screw material $\approx 200 \times 10^9$ N/m^2

$I_{\min}$ = minimum moment of inertia = $\frac{\pi d_0^4}{64}$

d_0 = diameter of lead screw

ℓ_k = the length of the lead screw, factored to account for the support conditions.

In the worse case, the lead screw is simply supported at each end:

ℓ_k = ℓ, where ℓ is the length of the lead screw

Table G6.1 shows a selection of available SKF planetary roller screws, with their load ratings and critical buckling length ℓ for an applied load of 15,000 N.

TABLE G6.1 DIMENSIONS FOR SKF PLANETARY ROLLER SCREWS

d_0 mm	P_h mm	D_{nut} mm	C_a kN	C_{oa} kN	ℓ, due to 15,000 N, mm
8	4	25	9.19	16.3	163
12	2	26	5.85	7.39	366
12	5	30	14.5	22.3	366
15	5	35	21.2	36.3	572

C_a - dynamic load rating, such that 90% of a large sample of screws are expected to attain or exceed 1 million revolutions under this constant, centrally acting pure axial load without fatigue (flaking).

C_{oa} - static load rating - that axial, constant, centrally acting load which produces a total permanent deformation of one raceway and roller of 0.0001 of the diameter of the curved surface of the roller.

P_h - pitch of screw (lead)

Fatigue of the lead screw will not be a problem as it will only be under load each time the inner cylinder has to be moved. A 12 mm diameter lead screw with 5 mm pitch should therefore be acceptable (this would allow a maximum stroke of 366 m) There must be sufficient control on the motor that the lead screw is only turned through 216° (equivalent to 3 mm movement).

G6.1 Frictional Force Required to Turn the Lead Screw

The input torque required to generate an axial load of 15,000 N needs to be calculated. The SKF catalogue gives the following equations to calculate this torque.

The torque $T = \dfrac{P_h\,(F + m_L g)}{2000\pi\eta_p}$

P_h	-	pitch (mm)
F	-	break-out friction force to be overcome
m_L	-	mass of inner cylinder and motor
g	-	acceleration due to gravity
η_p	-	real direct efficiency at converting rotary into linear motion.

m_L is small, hence $F >> m_L g$

$$\therefore\ T = \frac{P_h\,F}{2000\ \pi\ \eta_p} \qquad 6.1$$

The theoretical direct efficiency

$$\eta = \frac{\tan\phi}{\tan\,(\phi + \lambda)}$$

$$= \frac{1}{1 + \dfrac{\pi d_0}{P_h}\ \mu}$$

ϕ	-	helix angle of the screw thread
λ	-	friction angle
μ	-	coefficient of friction, which is a function of the helix angle

On average, $\eta_p = 0.9\eta$, although, because we are running at very slow speeds or for a very short time, the practical efficiency will be lower.

For $d_0 = 12$, $P_h = 5$, $\phi = 7.55°$.

From a graph in the catalogue, for $\phi = 7.55°$, $\mu = 0.012$

$$\therefore \eta = \frac{1}{1 + \pi\left(\frac{12}{5}\right)(0.012)}$$

$$\underline{\eta = 0.917}$$

$$\therefore \eta_p = 0.9 \times 0.917$$
$$\underline{\eta_p = 0.825}$$

Substituting η_p into equation 6.1, the required torque

$$T = \frac{5 \times 15000}{2000 \times \pi \times 0.825}$$

$$\underline{T = 14.5 \text{ Nm}}$$

Therefore, to generate an axial load of 15,000 N, the motor must provide 14.5 Nm. This will be at break-out.

For applications with high acceleration and/or high inertia, the starting torque may be several times greater. However, in this case inertia loads and speeds are slow so the starting torque should not be much higher than this. Ideally, the starting torque should be measured.

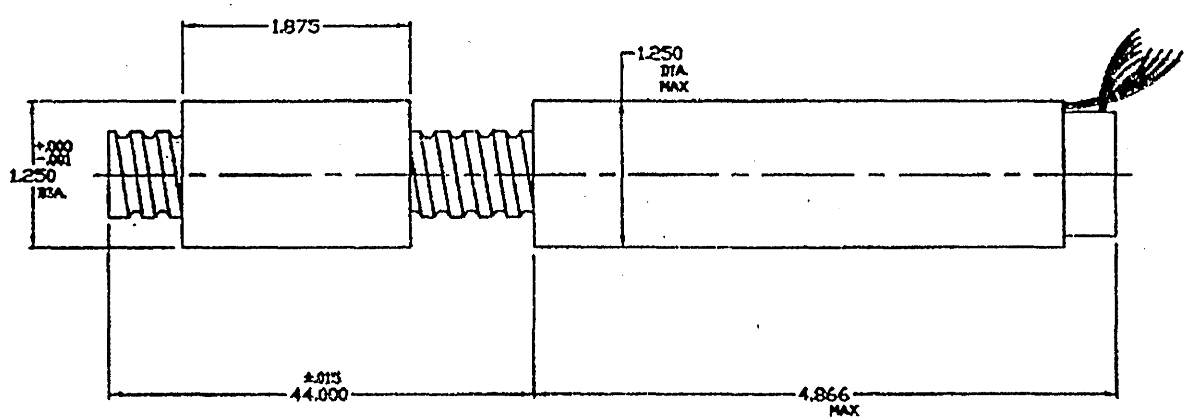
G7 MOTOR SELECTION

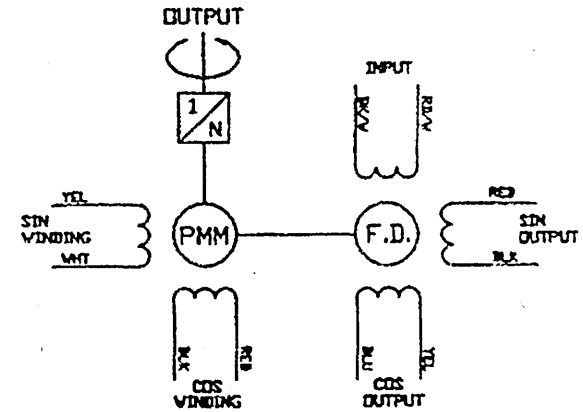
Kollmorgan Hitech supply special permanent magnet motors produced by Astro Developments in the USA (Ref 125). These motors have very high reliability and control, being used regularly to actuate control surfaces on aircraft. They have also been used inside oil wells and are therefore suitable for the harsh well environment.

Performance data for these motors is shown below. The motors include a high torque gearbox, brake and encoder.

The nozzle cuts on rotation and is traversed a distance equal to the nozzle diameter at the end of every cut. A smooth torque is required to rotate the nozzle over speeds from 10 to 200 mm/min. The translation can be at any speed.

NOTE: MINIMUM ENVELOPE SHOWN. MOUNTING AND NUT CONFIGURATION TO BE DETERMINED BY CUSTOMER





ROTATION CCW VIEWING SHAFT END

PINION DATA

No. OF TEETH	A/R
PITCH AT 20 DEGREE P.A.	
THEO. PITCH DIA.	
TESTING RADIUS	
OUTSIDE DIA. (REF.)	
INVOLUTE TOOTH FORM	
AGMA QUALITY No.	

PROPOSAL No. 6019B1

CUSTOMER	BHR CRNF
CUSTOMER SPECIFICATION	ASTRO PROP 6019
N.S.N.	

REV	DESCRIPTION	APPR
1	11-4-96 RELEASED -B1	

ASTRO INSTRUMENT CORPORATION

TITLE DOWN HOLE DRIVE

MOTOR DATA

PERFORMANCE AT 25°C UNIT TEMP

EXCITATION (VDC)	300
POWER OUTPUT AT 200 mm/min (WATTS)	26
OUTPUT FORCE AT 200 mm/min (N)	11,000

FIELD DIRECTOR DATA

PERFORMANCE AT 25°C UNIT TEMP

EXCITATION (VRMS)	10
FREQUENCY (kHz)	32
OUTPUT (VRMS)	10

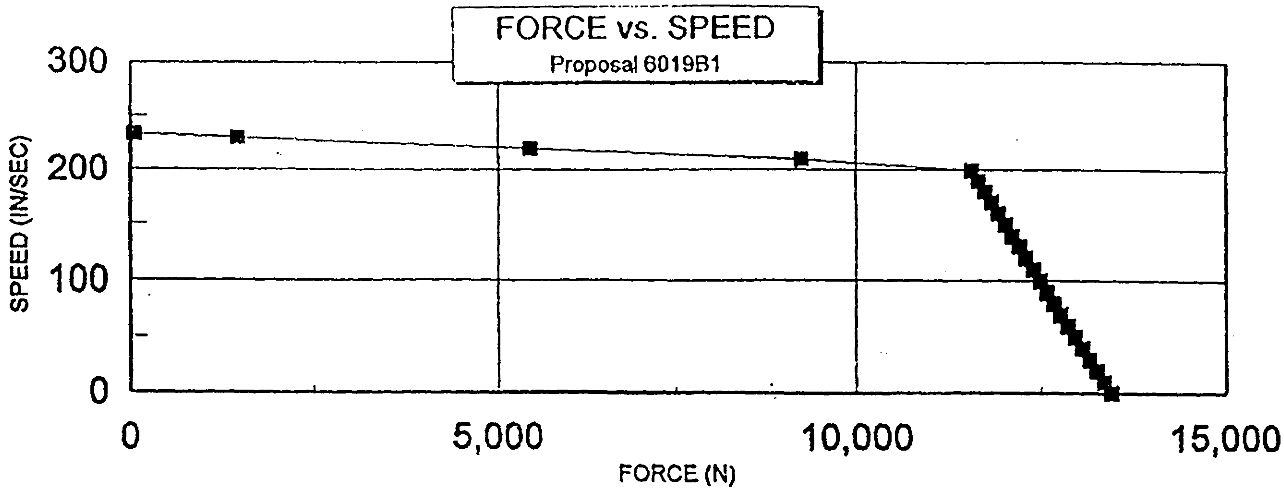
ADDITONAL DATA

UNIT OPERATING TEMPERATURE RANGE	-55 TO +250°C

UNIT CAPABLE OF OPERATION IMMERSED IN HYDRAULIC FLUID AT AN AMBIENT TEMPERATURE OF 200°C AND AT A PRESSURE OF 15,000 PSI

216, 0230

NOV 0 4 1996



MOTOR TYPE:	20		GHD=	216		
INPUT VOLTAGE=	270	*VDC*	DCR=	141	*OHMS*	
CURRENT LIMIT=	0.219	*ADC*	KT=	61400	*N/AMP*	
TEMPERATURE=	25	*°C*	KB=	6.62	*VP/IN/SEC*	
			LEAD=	0.230	INCHES/REV	

OUTPUT SPEED *(mm/min)*	INPUT VOLTAGE *(VDC)*	INPUT CURRENT *(ADC)*	POWER FACTOR *(DEG)*	INPUT POWER *(WATTS)*	OUTPUT FORCE *(N)*	OUTPUT POWER *(WATTS)*	POWER LOSS *(WATTS)*	EFF.
0	31	0.219	1.000	6.7	13,435	0.0	6.7	0.0%
10	42	0.219	0.998	9.3	13,338	2.2	7.0	24.0%
20	54	0.219	0.996	11.8	13,242	4.4	7.4	37.5%
30	66	0.219	0.993	14.3	13,146	6.6	7.7	46.0%
40	77	0.219	0.992	16.8	13,051	8.7	8.1	51.7%
50	89	0.219	0.990	19.3	12,956	10.8	8.5	55.9%
60	101	0.219	0.989	21.8	12,861	12.8	9.0	58.9%
70	113	0.219	0.988	24.3	12,766	14.9	9.5	61.1%
80	124	0.219	0.987	26.9	12,672	16.9	10.0	62.9%
90	136	0.219	0.986	29.4	12,578	18.8	10.5	64.2%
100	148	0.219	0.985	31.9	12,484	20.8	11.1	65.2%
110	160	0.219	0.985	34.4	12,391	22.7	11.7	66.0%
120	171	0.219	0.984	36.9	12,298	24.6	12.3	66.6%
130	183	0.219	0.984	39.4	12,205	26.4	13.0	67.0%
140	195	0.219	0.984	41.9	12,112	28.2	13.7	67.3%
150	207	0.219	0.983	44.4	12,020	30.0	14.4	67.5%
160	218	0.219	0.983	47.0	11,928	31.8	15.2	67.7%
170	230	0.219	0.983	49.5	11,837	33.5	16.0	67.7%
180	242	0.219	0.982	52.0	11,746	35.2	16.8	67.7%
190	254	0.219	0.982	54.5	11,655	36.9	17.6	67.7%
200	265	0.219	0.982	57.0	11,564	38.5	18.5	67.5%
210	270	0.183	0.987	48.6	9,251	32.3	16.3	66.5%
220	270	0.121	0.994	32.6	5,434	19.9	12.7	61.1%
230	270	0.058	0.998	15.5	1,464	5.6	9.9	36.1%
233	270	0.035	0.999	9.4	57	0.2	9.2	2.4%

11/04/96

G7.1 Summary of Required Motor Torque

Rotation	Minimum	Maximum
Seal break-out friction: measured to be	3.5 Nm	3.5 Nm
Bearing break-out friction: predicted to be	2.4 Nm	7.6 Nm
Total break-out Torque	5.9 Nm	11.1 Nm

Translation	Minimum	Maximum
Seal break-out friction: measured to be	1300	2600
Motor torque required to generate these axial loads (using Eqn. A5.1)	2 Nm	8.2 Nm

1.25" diameter motors with 1.25" diameter gearboxes are large enough to provide translational forces up to 11500 N at 200 mm/min and torques up to 12.5 Nm at very low speeds. They will therefore be ideal for moving the nozzle assembly.

G8 LOWER OUTER CYLINDER DESIGN

The top section of the lower outer cylinder will be identical to the upper cylinder with the seal ring cartridge held in place between the two parts of the outer cylinder.

Below this, the housing for the lead screw nut must be attached. This housing will use the M45 thread connection described in Section G1 with an 'O' ring and back-up ring.

Ideally the outer diameter of the 'O' ring should be less than the pressure diameter of 36.7 mm, calculated in Section G1. Also, the ID of the housing should be as large as possible to incorporate the motors and motor housing.

The closest standard 'O' ring and back-up ring available from James Walker has an OD of 37.8 mm and requires a recess diameter of 35 mm. The ID of the housing should then be 34 mm. This is shown in Figure G8.1. (This arrangement was used on the Seal Rig.)

Note the anti-rotation slots which are explained in Section 7.4.

For the pressure diameter to increase to 37.8 mm and the manipulator OD to be kept constant, the minimum material design stress must be increased:

$$f > \sqrt{3}\ P\ \frac{OD^2}{OD^2\ -\ ID^2}$$

$$f > \sqrt{3}\times 744\times 10^5 \times \frac{(53)^2}{(53)^2\ -\ (37.8)^2}$$

$$\underline{f > 262 \times 10^6\ \mathrm{N/m^2}}$$

This is an increase of 5%.

Therefore, the outer cylinder of the manipulator should have a minimum design stress at 200°C of 262 MN/m^2.

Tensile area check at screw thread:

Area of cross-section at male part = $\frac{\pi}{4}\left[(43.3\times 10^{-3})^2\ -\ (34\times 10^{-3})^2\right] = 5.6\times 10^{-4}\ m^2$

Area of cross-section at female part = $\frac{\pi}{4}\left[(53\times 10^{-3})^2\ -\ (45\times 10^{-3})^2\right] = 6\times 10^{-4}\ m^2$

Therefore the male part is the critical component.

The maximum applied load during normal operation
= pressure force over a diameter of 37.8 mm + friction force due to translating the inner cylinder

$= \frac{\pi}{4}\left[(37.8\times 10^{-3})^2\times 744\times 10^{-5}\right] + [15000]$

= 98492 N

≈ <u>100 kN</u> (as calculated in Section G1)

The maximum tensile load carried by the threads

$$= 5.6\times10^{-4}\times262\times10^{6}$$

$$= 146{,}720\ N$$

There is therefore a safety factor $= \dfrac{146{,}720}{100{,}000}$

$= \underline{1.46}$

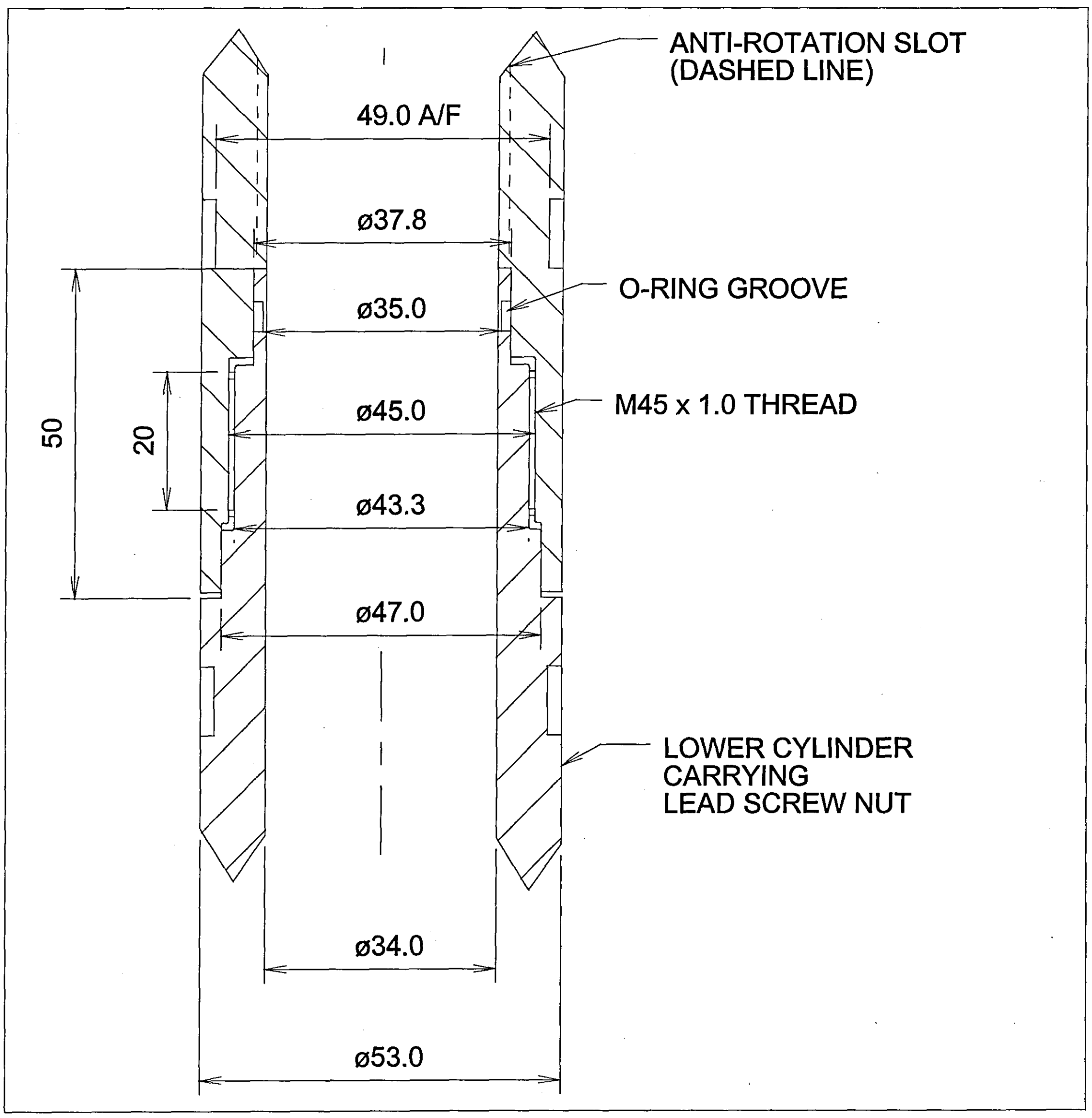


FIGURE G8.1 LOWER CYLINDER CONNECTION DESIGN

G8.1 Motor Housing Design

The OD of the motors is 1.25" (31.75 mm). The ID of the outer cylinder is 34 mm, although if the motors were housed in a female section of the outer cylinder, which does not carry the O-ring seal, this could be enlarged to 36 mm, which is just smaller then the pressure diameter calculated in Section G8 above.

A controller is required for the two motors, which sequences their motion and distributes power as required. A communication cable from the controller to the surface is needed to enable the operator to monitor the progress of the cut. Alternatively, the main controller could be at the surface, sending signals to the motors via the downhole controller.

The wireline cable, which carries the power cables and communication cables to the surface, must connect to a junction box at the top of the manipulator. Minimum diameter power and communication cables must then come from the junction box, pass through the inner cylinder to the motor controller.

Note that the cables must be protected as they pass through the inner cylinder. There must also be sufficient slack to allow them to stretch the 200 mm stroke of the lead screw and to rotate through an angle of up to 360°, as they pass through the motor/inner cylinder coupling. Careful monitoring of the rotation of the inner cylinder should ensure that the cables do not become entangled in the coupling.

The motors must be joined together using some form of connector, as shown in Figure G8.1.1, with the controller inside. The connector and motor housing wall thicknesses only need to be minimal (in the order of 1 to 2 mm) to transmit the 15000 N axial load and 11 Nm torque. A new housing design at the back of the motor (the encoder housing) will be required to do this, which should not be a problem.

A circular plate is fixed to the front of the rotational motor. This carries the anti-rotation pins, which locate in the 1.5 mm deep by 2 mm wide slots in the outer cylinder. The anti-rotation pins will be rectangular, and a minimum of 1.5 mm wide by 2 mm long. They should be covered by a low friction material such as PTFE.

The only problem with incorporating the motors is with passing the power and communication cables between the top motor housing and outer cylinder, a clearance of only 1.1 mm. The diameter of the cables is therefore critical.

The motor data sheets show that the required power supply is:

- 300 VDC
- 0.22 amps
- 57 Watts.

Standard cables are given in the RS Catalogue (Ref 126). The closest match for this application is called High Performance Flexlite. A 0.25 mm^2 cross sectional area cable is rated to 600 V and approximately 1 amp. The wire diameter is between 0.55 mm and 0.63 mm and the insulation diameter is between 0.91 mm and 1.04 mm. The cable is rated to 150°C, but this is only a function of the type of insulation. Other insulators should be available.

This cable is sufficiently small that it can be passed around the outside of the top motor. To prevent the possibility of interference between the cable and the outer cylinder, it should be attached to the motor housing and aligned to fit inside the anti-rotation slot. See Figure G8.1.1

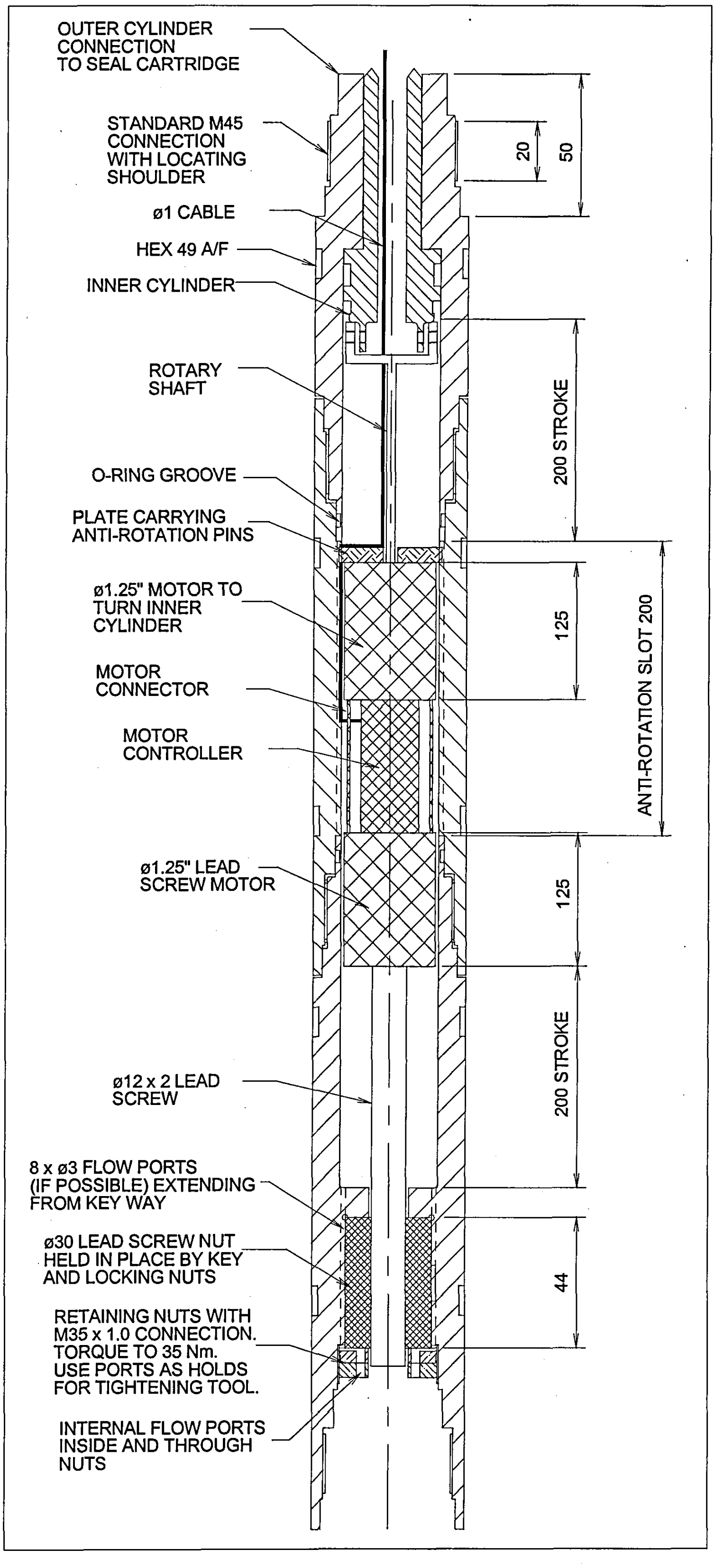


FIGURE G8.1.1 LOWER CYLINDER DESIGN INCORPORATING MOTORS, LEAD SCREW, LEAD SCREW NUT AND CABLE